P-1700

Performance Analysis of 210MW Steam Turbine at MTPS

A Project Report

Submitted by

M.S. Jayakumar - 71204407004

in partial fulfillment for the award of the degree
of

Master of Engineering
in
Energy Engineering

DEPARTMENT OF MECHANICAL ENGINEERING

KUMARAGURU COLLEGE OF TECHNOLOGY
COIMBATORE – 641 006

ANNA UNIVERSITY :: CHENNAI 600 025

APRIL– 2006

ANNA UNIVERSITY :: CHENNAI 600 025

BONAFIDE CERTIFICATE

Certified that this project report entitled "**Performance Analysis of 210 MW Steam Turbine at MTPS**" is the bonafide work of

Mr. M.S. Jayakumar - Register No. 71204407004

Who carried out the project work under my supervision.

Signature of the Head of the Department	Signature of the supervisor
Dr. T.P. Mani	**Mr.K.G.Maheswaran**
HEAD OF THE DEPARTMENT	SUPERVISOR

01/07/06.

Internal Examiner

Dr. T.P. Mani
B.E., M.E., Ph.D., DML., MIE., MNIOR., MISTE.,
Dean & HoD / Dept. of Mech. Engg.
Kumaraguru College of Technology
Coimbatore - 641 006

External Examiner

DEPARTMENT OF MECHANICAL ENGINEERING

KUMARAGURU COLLEGE OF TECHNOLOGY

COIMBATORE 641 006

TAMILNADU ELECTRICITY BOARD

O/o EXECUTIVE ENGINEER
T&E / MTPS / METTUR DAM – 6.

Thiru. M.S. Jayakumar, Final year M.E (Energy Engineering), Kumaraguru College of Technology, Coimbatore had done Project Work entitled **"Performance Analysis of 210 MW Steam Turbine at MTPS"** at Mettur Thermal Power Station during the period of 06.01.06 to 30.04.06.

M.S. Jayakumar

Specimen Signature
of the Student

EXECUTIVE ENGINEER
T&E / MTPS / METTUR DAM - 6

D.146
2.6.06

ANNAMALAI UNIVERSITY

DEPARTMENT OF MECHANICAL ENGINEERING

MECHSEM 2006

NATIONAL LEVEL TECHNICAL SYMPOSIUM

CERTIFICATE OF APPRECIATION

This certificate is awarded to Mr/Ms. M.S. JayaKumar

has presented a paper on PERFORMANCE AND ANALYSIS OF 210 MW STEAM TURBINE AT MTPS

in the MECHSEM 2006 organised by the Mechanical Engineering Department held on April 12th, 2006

Mr.L.VENKATACHALAM General Secretary B.E. (Mechanical) Final year	**T.SHANMUGAVADIVEL** Prof. of Mechanical Engineering Organising Secretary	**Dr.G.KAMARAJ** Prof. & Head Department of Mechanical Engg.

ABSTRACT

Title of this project work is **"PERFORMANCE ANALYSIS OF 210 MW STEAM TURBINE AT MTPS"**. This work is carried out at Thermal power station at Mettur in Salem district which is under control of Tamilnadu Electricity Board.

In this report the construction and internal structure of steam turbine, and its associated systems like condensing system, regenerative system, cooling system, circulating system etc are discussed in detail. Moreover the losses that are occurring in a steam turbine and its various parts like high pressure, intermediate pressure and low pressure parts of turbine and overall losses are calculated and due to this loss, the overall efficiency arrived in a full unit consists of a boiler, turbine, generator upto transformer are calculated. Further, the steps to be taken to reduce the loss and to conserve energy are discussed and by this way the performance improvement of turbine is discussed.

திட்டப்பணி சுருக்கம்

"மேட்டூர் அனல் மின் நிலையத்தில் உள்ள 210 மெகா வாட் மின்திறன் கொண்ட அனல் மின் சுழலியின் செயல்பாடும் ஆய்வும்" என்பதே இத்திட்டப்பணியின் தலைப்பாகும். இப்பணி சேலம் மாவட்டம் மேட்டூரிலுள்ள தமிழ்நாடு அரசு மின்சாரவாரியத்தின் கட்டுப்பாட்டில் இயங்கும் அனல்மின் நிலையத்தில் மேற்கொள்ளப்பட்டுள்ளது.

இந்த ஆய்வு கட்டுரையில் மின்சுழலியின் அமைப்பு பற்றியும், உள்ளே அமைந்துள்ள பாகங்கள் பற்றியும், செயல்படும் விதம்பற்றியும் அதனுடன் இணைந்து செயல்படுகின்ற நீராவியை குளிர்விக்கும் அமைப்பு, நீராவி சுழற்சி அமைப்பு, மீண்டும் சூடாக்கும் பயன் அமைப்பு போன்ற பல்வேறு அமைப்புகள், அதன் செயல்பாடுகள் குறித்தும் விரிவாக விளக்கப்பட்டுள்ளது மேலும் மின் சுழலியில் ஏற்படும் ஆற்றல் இழப்புகள், அதற்கான காரணங்கள் அதன் மூலம் ஒரு முழு அலகில் ஏற்படும் செயல்திறன் மாற்றங்கள் ஆகியவை இங்கு கணக்கிடப்பட்டு காட்டப்பட்டுள்ளது.

மேலும் இச்சுழலில் மற்றும் சார்பு அமைப்புகளில் ஏற்படும் ஆற்றல் இழப்புகளை குறைத்து, செயல்திறனை மேம்படுத்த மேற்கொள்ளப்பட வேண்டிய வழிமுறைகள் தரப்பட்டுள்ளன.

இதன் மூலம் மின் சுழலியின் சிறப்பான செயல்பாட்டை அடைய முடியும் என்பதை இந்த ஆய்வுக்கட்டுரை விளக்குகிறது. ஆற்றல் சேமிப்பு குறித்த விழிப்புணர்வு ஏற்பட்டுள்ள இச்சமயத்தில் இப்பணி பயனுள்ளதாக அமையும்.

ACKNOWLEDGEMENT

The Author wishes to thank his internal guide **Mr. K.G. MAHESWARAN,** Lecturer, Department of Mechanical Engineering, Kumaraguru College of Technology, Coimbatore for giving valuable guidance and continuous encouragement throughout this project.

The author sincerely thanks to **Dr. T.P. MANI,** Head of the Department, Department of Mechanical Engineering, Kumaraguru College of Technology, Coimbatore for giving permission and valuable suggestions to complete this work.

The author thanks to The Principal **Dr. K.K. PADMANABHAN,** Kumaraguru College of Technology, Coimbatore who allowed us to do this project and completing my course.

The author gives immense pleasure to express thanks to **Er. S. MANIMARAN,** Asst. Engineer, Operation and Efficiency, for his kind and continuous help and valuable guidance in completing my project.

The Author thanks to **Er. G.RAMADOSS,** Asst, Engineer, Technical Service, **Er, V. RAMAKRISHNAN,** Asst Engineer, Turbine Maintenance (UCB), Mettur Thermal Power station, Mettur, for his valuable guidance and encouragement.

The Author express thanks to **Dr.V.Velmurugan** & **Mr.S.R.Rajabalayanan** , Dept. of Mechanical Engineering , Kumaraguru College of Technology for their kind help and support for completion of project report.

Last but not least author wish to thank my beloved parents and friends who give me support and encouragement in completing this project.

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SYMBOLS & ABBREVIATIONS

Amps	:	Ampere
C	:	Centigrade
CRH	:	Cold Re Heat
CV	:	Control Valves
ESV	:	Emergency Stop Valve
GC	:	Gland Cooler
HP	:	High Pressure
HPT	:	High Pressure Turbine
HRH	:	Hot Re Heat
IP	:	Intermediate Pressure
IPT	:	Intermediate Pressure Turbine
Kcal	:	Kilo Calories
KJ	:	Kilo Jules
KSC	:	Kilo Gram / Sq.Cm
KV	:	Kilo Volt
KW	:	Kilo watt
KWH	:	Kilo Watt Hour
LP	:	Low Pressure
LPT	:	Low Pressure Turbine
MTPS	:	Mettur Thermal Power Station
MW	:	Mega Watt
RPM	:	Revolution Per Minute
t	:	Tones
T / Hr	:	Tones Per Hour

CHAPTER – 1

METTUR THERMAL POWER STATION

1.1 INTRODUCTION

India, which had power generation capacity of only 1350 MW during its independence, has made significant growth in this vital infrastructure sector by raising its capacity currently to approximately 1,06,000 MW.

Major share of India's power generation has been contributed by 210 and 500 MW Thermal power stations. Hence in this increasing population, the energy demand is also increasing, so its important to produce power in the power stations and supply without break down or obstacles. For this the performance monitoring and analysis of the power stations and their parts are essential. By this performance analysis, the problems that are associated with proper power production and distribution can be found out and it will help to improve the operation and maintenance in a good way. So it is important to undergo this study.

The scope of this work is to have a clear vision on operation of the system, say 210MW steam turbine in Mettur thermal power station, and its performance variation by observing the losses especially heat energy loss.

By estimating the heat loss that are occurring in this turbine, the ways to reduce the losses and to improve the performance and there by achieving the higher efficiency can be found out. Hence in this work, the working of the system is discussed and its heat losses at various stages of turbine and overall head loss are calculated and overall efficiency of a unit which consists of a boiler, turbo-generator is estimated.

In this report, the brief discussion about the thermal station and the turbine and its various parts and various auxilliaries and systems associated are discussed and then the loss calculation is given which is estimated by the calculation of heat losses at several extractions and various parts of turbine, then finally the performance monitoring and suggestion to reduce the loss as improve the performance are explained.

The estimation of heat losses that are occurring deep inside the parts of turbine will be difficult since the higher pressure and temperature of around 530°C that are available inside the turbine. so more accurate heat loss estimation is limited due to non availability of measuring equipments at that condition.

The performance of turbine is monitored and recorded in a separate section called operation and maintenance in MTPS where the replacement of spares, performance variation before and after replacement and mainly the load current difference is estimated and thereby cost saving is calculated. But this attempt of performance analysis and heat losses calculation associated with the operation of turbine is a new work in MTPS. By this work the losses in turbine can be viewed and ways to reduce the loss and improve the performance can be taken into account. The over view of Mettur Thermal power station is shown in fig.1.1

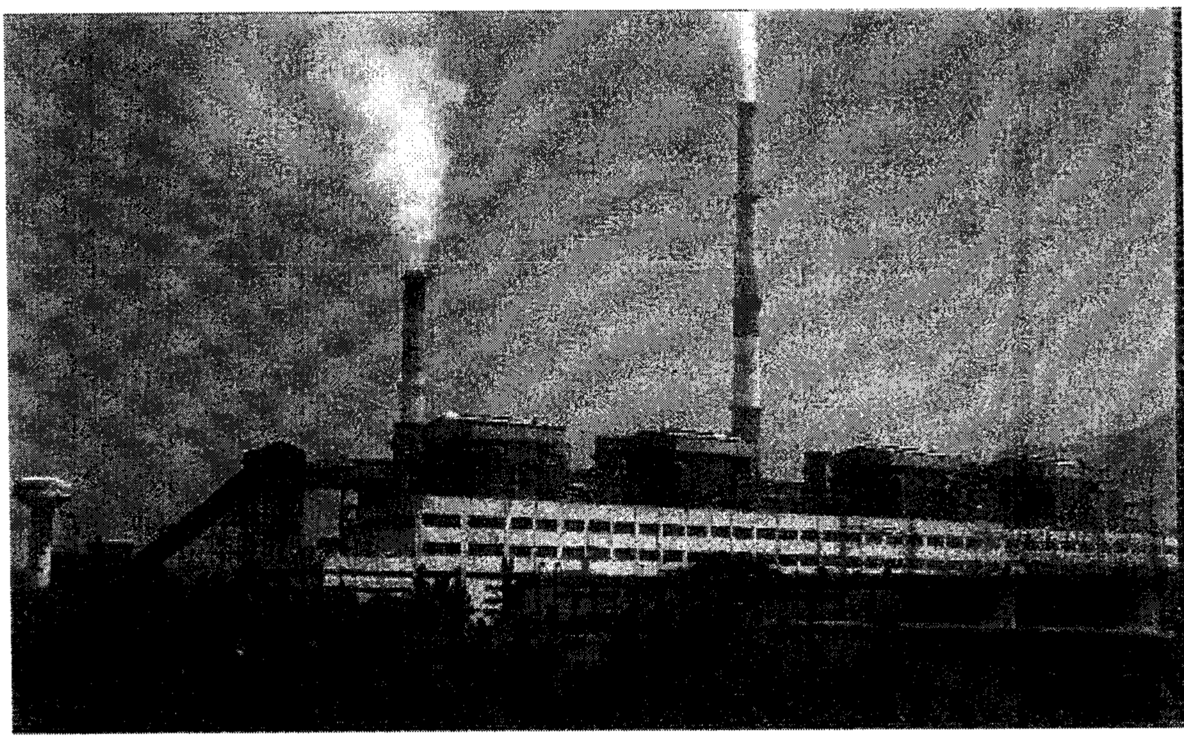


FIGURE 1.1 METTUR THERMAL POWER STATION

1.2 INTRODUCTION TO METTUR THERMAL POWER STATION

Mettur Thermal Power Station is the one and only inland thermal power station of Tamil Nadu Electricity Board. The station is situated in the left flank of the Ellis Surplus course of the Stanley Reservoir, Mettur Dam. The Main objective of the installation of Mettur Thermal Power Station with 4 Nos. 210 MW units, is to cater to the need of industrial centres of the state of Tamil Nadu, ie. Salem, Erode and Coimbatore. The project work commenced in the year 1981 and the first unit was synchronized with the grid in 1987 and the fourth unit in 1990.

Mettur Thermal Power Station is getting its coal supply from Mahanadhi Coal Fields, Orissa, Coal from Paradeep Port, Orissa is transported through ship to Ennore port and from there by rail to Mettur. Four Wagon tipplers have been erected to tip the coal wagons. The coal is fed into the coal bunkers through mechanical conveyor system. There are two streams of coal conveyors with 1400 Tonnes / Hour capacity each. There is a coal yard with space to stack 4.0 lakh tones of coal, which is sufficient to meet one month's requirement. There are two stacker-cum-reclaimers in the yard to stack the crushed coal and to retrieve the coal to bunkers when necessary.

A dyke has been constructed across the perumpallam valley over an area of 1268 acres. The dyke consists of Upper Ash Dyke, Lower Ash Dyke and Two settling ponds in series. The ash slurry is pumped into the dyke. Ash settles down and the water flows into the primary pond and then the secondary pond from where clear water is let into the river Cauveri. The effluent water is tested for very high degree of purity in accordance with the standards fixed by TNPCB.

To augment the capacity of the upper ash dyke, it has been proposed to acquire land in the area up to +280 metre level above MSL. In this process, about 358 families of Pudureddiyur and Puduchinnakkavoor are being shifted to an alternate site. The G.O. for the acquisition has been issued and the acquisition is under processing. After the acquisition of the alternate site, 358 families will be allotted with 3 cents of land for construction of building.

The power generated at 15.75 KV is stepped up to 230 KV and fed into the Tamil Nadu Grid. The power generated in the station is being evacuated into the gird by 8 Nos. 230 K.V. feeders.

1. MTPS – Mettur 230 KV Auto Substation feeder.
2. MTPS – Ingur 230 KV Substation feeder.
3. MTPS- Salem 230 KV Substation feeder.
4. MTPS – Salem 400 KV Substation Feeder – I.
5. MTPS – Salem 400 KV Substation Feeder – II.
6. MTPS – Singarapet 230 KV Substation feeder.
7. MTPS – Gobi 230 KV Substation feeder.
8. MTPS – Mettur Tunnel Power House feeder.

Mettur Thermal Power Station has secured the eligibility for Gold Medal from Govt. of India for the fifth consecutive year commencing from 2000-2001. The station is also winning awards from Govt. of India for reduction in secondary fuel oil consumption.

1.3 SALIENT FEATURES

1.	Location	:	Mettur (11° 56' North Latitude and 77° 48' East Longitude – Elevation 123 metre above MSL)	
2.	Area of the Plant	:	1. Main Plant	: 362 Acres
			2. Railway siding	: 30 Acres
			3. Raw Water P.H And storing shed	: 73 Acres
			4. Ash dyke I stage	: 781 Acres
			5. Ash dyke II stage	: 487 Acres
			Total	: 1733 Acres
3.	Cost of the Project	:	I Stage	: Rs.384.30 Crores
			II Stage	: Rs.351.76 Crores
			Total	: Rs.736.06 Crores
4.	Capacity of Unit	:	I Stage	: 2 × 210 MW
			II Stage	: 2 × 210 MW

5. Probable Generation p.a	:	I Stage : 2247 M.U. II Stage : 2247 M.U.
6. Requirement of fuel (coal)	:	14000 Tonnes / Day for 4 Units
7. Requirement of Water	:	70 cu. Secs.
8. Chimney Height	:	I Stage – 130 Metres II Stage – 220 Metres
9. Main Parameters		
i. Boiler Capacity	:	700 Tonnes of Steam / Hour
ii. Temperature of Steam	:	540° C
iii. Pressure of Steam	:	137 Kg/cm^2
iv. Turbine	:	3 stage turbine
v. Generation Voltage	:	15.75 K.V.
vi. Generation Capacity	:	210 M.W.
10. Coal conveyor System		
a. Capacity	:	2800T/Hr.(Two streams of 1400T/Hr)
b. Stacking Capacity	:	4 lakhs tones (Sufficient to feed 4 Unit for 30 days)
Cooling Water requirement		
a. Raw Water	:	3800 M^3 / Hr for each stage
b. Cooling Water	:	3200 M^3/Hr for each Unit
11. Station Transformers	:	2 × 31.5 MVA, 230/7 KV
12. 230 KV Feeders	:	Total – 8 Number
13. H.F.O. Tank Capacity	:	1 × 70 K.L.
14. H.S.D. Oil Tank capacity	:	1 × 70 K.L
15. Computer	:	* Data Acquisition System with Hitachi Model 80-M Computer for Unit-1. * Distributed Digital Control System with TATA HONEY WELL Ltd for Unit-2. * Distributed Digital Control System with Hitachi Computer for Unit-3 &4.

1.4 A PROFILE

Mettur, situated 11°56' North Latitude and 77°48' East Longitude, is a big industrial area in Salem District of Tamil Nadu. The place is famous for the Dam, constructed across the river Cauveri during the year 1928. Two hydel power stations of capacity 40 MW and 200 MW are already in existence. However, to even out the fluctuations in generation due to the irrigation based generating stations and to augment the generating capacity of Tamil Nadu grid, Mettur Thermal Power Station was proposed at Mettur where Cauveri water is available for its use.

This is the first inland thermal power station of Tamil Nadu Electricity Soard. The plant is located on the left flank of the Ellis surplus course of Mettur Reservoir. It is situated at an altitude of 213 Mts, above MSL. The construction work of the project commenced on 21.12.19~' The first unit was commissioned on 7.1.87 and the other 3 units were commissioned on 1.12.1987,22.3.1989 and 27.3.1990 respectively.

1.5 MAIN EQUIPMENTS

1.5.1 Boiler

The boilers of the Mettur thermal Power Station, capable of producing 700 tonnes of steam per hour at 540°C and 137 Kg/Cm2 pressure. While the boilers of the first two units were capable of working with a maximum of 60% oil firing, the boilers of the second stage are designed for 100% oil firing capacity.

1.5.2Turbo Generator

The Turbo Generators were supplied by MIs. SHEL, Hardwar. The turbine is a three stage turbine and the generator is capable of generating 210 M.W. at a generation voltage of 15.75 K.V.

1.5.3 Instrumentation and Control

A microprocessor based furnace safeguard supervisory system is installed for the first time at Mettur. The main instrumentation and control system envisages automatic control of the various parameters like pressure, temperature, flow etc., For the second stage the "Distributed Digital Control" system has been installed for the first time in Southern Region along with the regular "Data Acquisition System".

By this system, automatic computerised operation is carried out from the operator control desk provided with colour visual screens.

1.6 SPECIAL FEATURES

The Mettur Thermal Power station has the following special features.

- Tallest Chimney of 220m Height to take care of environmental and ecological needs.
- Microprocessor based furnace safe guard supervisory system and
- Distributed Digital Control System.

1.6.1 Coal Linkage

Mettur Thermal Power Station is getting its coal supply from Raniganj and Mugma collieries in Bengal/Bihar Area and Talcher Collieries in Orissa State. Although, the coal linkage for the station is from Singareni collieries in Andra pradesh, Coal from Singareni is not being received. Now coal from far away Orissa coal fields is received by rail cum sea cum rail route, paying high cost for transportation in the country. The daily requirement of coal for all the 4 units of 210 MW each is about 14,000 M.T. per day.

1.6.2 Wagon Tippler

4 Nos. Wagon tipplers have been erected and these are capable of handling the coal received in railway wagons. These tipple coal wagons and unload the coal of about 58 tonnes per tipping into the hoppers. The coal is fed through mechanical conveyor system of capacity of 2800 tonnes/hr of coal. Two strems of 1400 Tonnes/hr. are available. Space has been provided for stacking 4 lakhs tonnes of coal, which will be sufficient to feed all 4 units for 30 days. Two stacker-cum-reclaimers capable of stacking the crushed coal in the coal yard and to reclaim to feed the boilers have also been commissioned.

1.6.3 Ash Disposal

A dyke has been constructed across the perumpallam valley which consists of two settling ponds in series. The ash in the slurry pumped into the big pond at higher elevation settles down and partially clear water flows by gravity to the second pond. After further settlement of ash, clear water is let into the river Cauveri, The w~ter, let into the river is tested for very high degree of purity in accordance with the standards fixed by the Environmental and Pollution Control Board.

1.6.4 Evacuation of Power

The power generated at 15.75 KV is stepped up to 230 KV by 3 phase 250 MVA transformer (one for each unit) and fed into the Tamil Nadu grid. The station has been connected to T.N.E.B. grid by 8 Nos. 230 K.V. feeders to evacuate the power generated by the four units.

CHAPTER - 2
BASIC PRINCIPLES

The thermal power plants with steam turbine uses Rankine cycle is a vapour power cycle having two basic characteristics:

i) The working fluid is a condensable vapour which is in liquid phase during part of the cycle and

ii) The cycle consists of a succession of steady flow processes, with each processes, with each process carried out in a separate component specially designed for the purpose. Each constitute an open system, and all the components are connected in series s that as the fluid circulates through the power plant each fluid element passes through a cycle of mechanical and thermodynamic stages.

2.1 TEMPERATURE ENTROPY DIAGRAM

The temperature – entropy (T-S) diagram is probably the most useful diagram of al for illustrating certain fundamental point about Rankine steam cycles. Ideal condition for a unit on a T.S diagram are indicated in fig. , The unit uses steam at a pressure of 100 bar absolute, temp. 540^0C (813^0K) and rejects it to the condenser at 30m bar (saturation temp 24.1^0C).

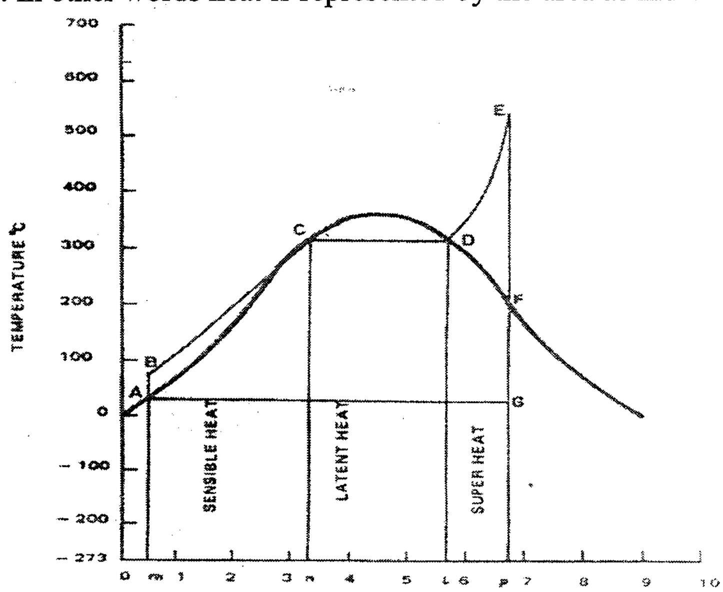
At point 'A' the condensate is at boiling temperature corresponding to the back (condenser) pressure. Its pressure is raised to 100 BAR in Feed Pump corresponding to point 'B'. Heat (sensible) is added to this water to raise its temperature. At the point C it reaches its saturation temp. at a pressure of 100 bar. Evaporation begins at the point. C Heat (latent- because no rise in temperature between C and D, as evident from the diagram), addition continues. At D all the water evaporates and super-heating commences. This is shown by the curve DE.

Steam then expands isentropically i.e. enters the turbine and rotates it, as shown y the line EFG. At point F there is not superheat left in the steam and so from F to G there is increasing wetness. At G steam is at a pressure of 30m bar and is passed out of the turbine to the condenser and condensation of steam takes place as represented by the line GA. At point A the steam has all been condensed and condensate is at boiling temperature read y to begin another cycle.

To summarise the above

AB	Pressure Rise in BFpp.
BC	heating of feed water (i.e. sensible heat addition)
CD	evaporation of water in boiler (i.el. latent heat addition)
DE	superheating of steam (i.e. superheat addition)
EFG	expansion pf steam in turbine, point E denotes demarcation between superheated and wet steam.
GA	condensation of steam in the condenser.

An important basic fact to remember is that heat is product of absolute temperature rand change of entropy. In other words heat is represented by the area as indicated in fig 2.1



Entropy

FIGURE 2.1 TS DIAGRAM

CHAPTER - 3

TURBINE DESCRIPTION

3.1 GENERAL DESCRIPTION OF A 210 MW (LMW) STEAM TURBINE

Before discussion in details about various features of the steam turbine and its auxiliaries let us have an overview of the system as a whole. fig. shown the schematic of a 210 MW steam turbine (BHEL/LMW).

Superheats steam (130 Kg/cm^2, 535^0C) from the boiler enters into the high pressure turbine through two emergency stop values (ESVs) and four control valves (CVs).

The high pressure turbine (HPT) comprises of 12 stages, the first stage being governing stage. The steam flow in high pressure turbine (HPT) being in reverse direction, the blades in high pressure turbine HPT are designed for anticlockwise rotation, when viewed in the direction of steam flow.

After passing through high pressure turbine (HPT) steam (27 Kg/cm^2, 327^0C_2) flows to boiler for reheating and pressure turbine (IPT) through two interceptor values (IVs) and four control valves (CVs) mounted on the IPT itself.

The intermediate pressure turbine has 11 stages. High pressure turbine (HPT) and intermediate pressure turbine (IPT) rotors are connected by rigid coupling and have a common bearing.

After flowing through intermediate pressure turbine (IPT), steam enters the middle part of low pressure turbine (LPT) through tow crossover pipes. In low pressure turbine, steam flows in the opposite paths having four stages in each path.

After leaving the low pressure turbine the exhaust steam (0.09 Kg/cm^2 abs) condenses in the condensers welded directly to the exhaust part of the low pressure turbine.

Rotors or intermediate and low pressure turbiners are connected by a semi flexible coupling.

The direction of rotation of the rotors is clock wise when viewed from the front bearing and towards the generator. The three rotors are supported in five bearings. The common bearing of high pressure and intermediate pressure rotors is a combined journal and radial thrust bearing.

Turbine is equipped with a turning (barring) gear which rotates the rotor of the turbine at a speed of nearly 3.4 rpm for providing uniform heating during starting and uniform cooling during shut down. Seven steam extractions for feed water heating have been taken form 9^{th}, 12^{th}, 15^{th}, 18^{th}, 21^{st}, 23^{rd}, & 25^{th} stage.

Condensate from the hot well of condenser is pumped by the condensate pumps, and supplied to the deaerator through ejectors, gland steam cooler and four number low pressure heaters. Steam extracted from various points of the turbine to heat the condensate in these heat exchanges. Form deaerator the feed waster is supplied to boiler by boiler feed pumps through three number high pressure heaters.

3.2 SPECIFICATION OF 210 MW (LMW) TURBINE

3.2.1 Turbine main data

Rated output of turbine	210 M.W.
Economic load turbine	202 M.W
Rated speed	3000 RPM
Rated pressure of steam before ESV	130 kg/cm^2
Rated live steam temperature	535^{0}C
Rated reheated steam pressure/ temperature temperature before IV.	24.49 kg/cm^2/535^{0}C
Steam flow at VWO condition	670 t/h
Steam flow at 210 MW	652 t/h
Steam flow through I.P.T	565 t/h (approx.)
Steam flow through L.P.T. condenser	450 t/h (approx.)
Rate pressure at L.P.T. exhaust	65.4 mm of Hg.
Rated C.W. inlet temperature	27^{0}C
Rated C.W. flow through condenser	27,000 m^3/h

Specific heat rate of turbine	2040 kCal/kWh
Rated efficiency of turbine	42.15%
Stages of turbine H.P., I.P. & L.P.	12, 11, 8 (4 x2)
No. of bearings for turbine	Five (second thrust)
Condenser specifications	14, 600m, 2 pass, 15, 620 tubes 30 mm dia, 10 m long.
Extraction tappings	25^{th}, 23^{rd}, 12^{th}, 9^{th}, Stage (LP 1 to HP 7).
Overall thermal expansion of HP turbine	32 mm (Approx.) From
Overall thermal expansion of LP turbine	15 mm (Approx.) anchor point of
Operating lubricating oil pressure/temp.	1 Kg/cm^2/40-45^{0}C
Operating governing oil pressure	20 kg/cm
Barring gear specifications	3.4 RPM 220:1 gear ratio.
Extraction pressure flow	HP 7-42kg/cm ; 34 t/h ; HP 6-26kg/cm ; 45 t/h ; HP 5-13kg/cm ; 17 - 18 t/h ; HP 4-7kg/cm ; 22 t/h ; LP 3-3.5kg/cm^2 ; 25 t/h ; LP 2-1.4kg/cm^2 ; 26 t/h ; LP 1-vac., 13 t/h ;
Differential expansion limits	HPT -+4 mm or (-) 1.2 mm IPT - +3 mm or (-)2.5 mm LPT -+4.5 mm or (-)2.5
Opening of values with reference to speeder gear position	IV – Start at 4.6 mm, full open-6.0 mm ESV-Start at 5.5 mm, full open – 6.7 mm CVSM – Start at 9.6 mm, full open – 12.1mm
Start of opening of HPT. IPT, CVs with reference	1 2 3 4 HPT - 0 0 72 90

to cam angle	IPT - 5 6 44 79
Condensate flow through Main ejector/GC-I/GC-II	ME-125 t/h GC-1-420 t/h and GC-II-260 t/h
First stage pressure/temp.	92 kg/cm /510-515^{0}C
Turbine seal steam pressure/temp.	0.2 kg/cm / 15^{0}C
Stator water flow/pressure	27m /2.4 kg/cm
Capacity of HP/LP bypass	30%
Specification of oil	Turbine 14, sp. gr. 0.852 at 50^{0}C
Excitation	static excitation system
I.S.P.U.G	1117 kg/cm^2 (starts unloading)
Bearing temp. maximum limit	90^{0}C (alarm 75^{0}C)
Turbine gland sealing Chambers	HP-F/R-5/4 Nos. IP-F/R-4/3 Nos. LP – 2/2 Nos.
Critical speeds	1585, 1881, 2017, 2489 4500 rpm.
Over speed trippings	10%, 11%, 16%

3.3 GENERATOR CONSTRUCTIONAL DETAILS

The large capacity Generators that are coupled to the steam turbines spin at 3000 R.P.M. to convert the mechanical power steam Turbine into Electrical power. The excitation wind to create the required magnetic field is provided in the rotor. The electrical power is generated by the interaction of the fluxes at the air gap-one due to the magnetic field, created the excitation current supplied to the rotor and the other due the armature reaction flux created by current in stator of the generator.

The mechanical power is converted at the air gap in electrical power. The conversion of power takes place wind certain amount of losses like windage losses, hysteresis losses frictional losses, and copper losses. The huge amount of power which is generated at the generators calls for a robu construction of the generators. As the forces that come on to generators are of mechanical, electrical, and thermal due ca has to be taken in the selection of the material that go in the design and construction of the generators.

The modern design takes in to consideration technological developments that have taken place in the metallurgy, insulation, electronic and chemistry to build generators of optimum size and with high efficiency.

The generator consists of the following parts.

1. Frame, Core, Coils, Insulation and Gas collers.
2. Rotor- solid rotor, Coils, Insulation, endring and Slip ring, Vide Fig 1,2 and 3.

In addition to the above the generators have been provide with a control system to control the excitation, a protection system to protect it from reaching over spead, protection against internal and external faults. It has a sophisticated in built arrangement to seal the hydrogen which is used for cooling windings as it is a highly explosive gas.

The constructional details of the stator and the rotor are as follows.

3.3.1 Rotor Body and Shaft

The rotor must carry the excitation winding, and should have a low reluctance path and transfer the rated torque from the turbine to the electromagnetic reaction at the air gap.

Steel is the only metal which meets these requirements. The typical alloy constitutions are as follows.

2.5% Nickel	1.2% Chrome	0.2% Vanadium
0.65% Manganese	0.5% Molybdenum	
0.25% Carbon	0.25% Silica	

The following are the design consideration for choosing the materials for, the construction for the rotor.

a. The more the ampere turns the rotor can carry the smaller the generator becomes together with the need for keeping a low current to minimize the losses and temperature, as much of the area as possible must be allocated to the winding copper.

b. The winding insulation must be mechanically strong to withstand centrifugal and bending stress, and stable to withstand load cycling. Adequate electrical tracking distances from the winding to the rotor body must be provided, since the winding is in direct contact with the ventilating gas, and dirt and off may collect on insulation surfaces. The insulation must therefore more substantial than the operating voltage of about 600 V required in other applications.

c. Passage for an appropriate flow of cooling gas must be provided in the coper section, and also in the steel body section to ensure that the specific temperature rise is not exceeded.

d. The magnetic flux in the rotor is unidirectional and normally substantially constant, so there is no loss due to magnetic hysteresis or eddy currents, but there must be adequate magnetic section particularly in the area of the winding slots. A high degree of saturation in the teeth would result unacceptably high excitation requirements and losses.

e. The whole centrifugal force of the slot contents retaining wedge and the tooth is resisted by the narrow section of the tooth, normally the tooth root. There must be an adequate safety margin between the maximum tooth stress at over speed and proof stress of the steel.

The optimization of these conflicting requirements has led in the latest designs, to a departure from parallel sided slots to trapezoidal slots.

During three phase sudden 'short circuits at generator terminals, torque peaks of four to five times full-load torque are experienced between L.P. turbine and generator shafts. The generator rotor shaft and coupling at turbine end must designed to withstand this peak torque. The coupling is usually shrunk on or keyed or dwelled, and has oil injection grooves for removal.

3.3.2 Rotor Winding

Winding coils are assembled into pairs of rotor slots symmetrically disposed about the pole axis. Because of the rotor winding slots are cut radially, it is not possible to fit a preformed coil in to the slots. Each turn is therefore assembled separately, either as half-turns or more pieces, with joints either at the centers of the end turns or at the corners, being brazed together after each turn is assembled, to form a series connected coil. Hard, high conductivity copper, with a small silver content to improve its creep properties, it used for the coils.

The coils are not individually ", rapped with insulation. Instead slot liners of molded glass fiber', or a composite of glass fiber and a more flexible insulating, material, insulate the coils from sides and bottoms of the slots,' and block of insulation separates the top turn from the wedge. Between each turn, thin separators of glass fiber or similar material, serve to insulate against the 10V or so between turns. Thick layers of insulation material on inside surfaces of, the-end rings and end disc insulate them from the end windings: The space between turns in the end are partially filled which insulating blocks, which ensure that the coils do not distort and which contain holes und passages for transfer of ventilating gases.

The high rotational speed produces a pressure head through the rotor slots which cause hydrogen to flow from both ends, under end windings and axially, through subs lots in the rotor and channels in the coils, whence it emerges radially through wedges into the air gap.

Fan mounted on the rotor primarily to circulate hydrogen through the stator, assist in the natural flow through the rotor.

The ends of the windings are connected to flexible leads, made from many thin copper strips, which run radially inward on to the shaft at the exciter end.

3.3.3 Rotor End Rings

Thick end rings are used to retain the rotor end winding from flying out under the action of the centrifugal forces. For electromagnetic reasons, the rings have traditionally been made from non-magnetic steel typically a 185 Mn, 45 Cr austenic steel. A 0.2 proof stress of 1000 MN./Square is available to cope with the high operating stress.

The end ring is prevented from moving axially either by means of lugs. Mating with similar lugs on the rotor body or by small spring loaded plungers located in to grooves.

The must be heated to about 300 degree centigrade to expand it sufficiently for the shrink surface to pass over the mating area of the rotor. The heat is applied by a special cylindrical electrical heaters.

3.3.4. Wedges and Dampers

The winding slot contents are retained by wedges they are non-magnetic in order to minimize flux leakage around the rotor circumference. The wedges should contain holes or slots through hydrogen should pass. These wedges are of extruded aluminum they act as damper winding during short circuits when the negative phase sequence flows in the rotor.

3.3.5. Slip Rings, Brush Gear and Shaft Earthing

Connections are taken from the D-leads in the bore, through radial copper connections and flexible connections on to slip rings. The slip rings must have a large surface area and run cool in order to transfer this current satisfactorily. One design uses two slip rings of the same polarity in parallel.

The brush gear is arranged with several brushes and holders on one or several removal brackets, each of which can be withdrawn for brush replacement while running on load. Brush pressure is maintained by constant pressure springs.

It is normal for a larger generators to produce an no-load voltage of 10-15 V between its two shaft ends, due to magnetic dissymmetry and other causes. This voltage drive current axially through the rotor body, returning through bearing and journals, causing damage to the surfaces. To prevent his insulation barriers are provided at the exciter end.

While all the insulation remain clean and intact, a voltage will exist between the shaft of the exciter end and earth, and this provides another method of the exciter end and earth, and this provides another method of confirming the integrity of the insulation. A shaft riding brush enables this shaft voltage to be monitored and an alarm is initiated when this falls below a predetermined valve.

It is important that the shaft at the turbine end of the generator is maintained at earth potential and a pair of shaft riding brushes connected to the earth to achieve this.

3.3.6 . Fans

Fans circulate hydrogen through the stator and the coolers. Identical fans are mounted at each end of the shaft, each ventilating half the axial length of the generator.

3.3.7. Bearing and Seals

The turbine end bearing is situated in a common pedestal with LP turbine outboard bearing the exciter – end bearing is either located in the end shield or in a separate pedestal. The white metal bearings are spherically seated for ease of alignment, are pressure lubricated and are provided with jack oil tappings. Seals are provided in both endshields to prevent the escape of hydrogen along the shaft. Most of these seals are of small thrust bearings, in which a non-rotating white-metal ted ring bears against a color on the shaft.

3.4. THE STATOR

3.4.1. Stator Core

The core provides path for the magnetic flux from the rotor pole around the outside of the stator winding and back in to the other pole.

As the rotor rotates, carrying its flux distribution with it, all points in the stator core experience a sinusoidally varying 50HZ flux density. This would indusce a 50HZ voltage of about 700 V axially in a solid core and to prevent large circulating currents with associated losses, the core is made of thin steel plates coated with a insulating material. The voltage induced axially in each plate is about 50 mV.

Core plates are cut to form segments of annular ring, twelve segments per ring is common. Winding slots location notches and holes for ventilation are cut in pressing operation.

With the core frame axis vertical and one core end plate in position at the lower end of the frame, a ring of core plates is assembled, located on dove-tailed keys on the inside periphery of the frame. The radial butt joint between plates has a small a gap as possible to minimize magnetic flux distortion. The next ring of core plates is assembled so that its butt joints do not coincide with those of adjacent rings.

Gaps in the build-up of core plates are created where required, for passing of cool gases, by bilding in a ring of thick plates to which small steel bars have been welded. These bars are aligned in a mainly radial orientation and serve to distribute the gas through the ducts. Holes in the plates are arranged to be in axial alignment and thus form axial ventilation ducts in some designs. At intervals during core buildings, heavy pressure is applied to consolidate the assembly of plates.

When the build is almost complete, and with pressure applied at the top end, the core is subject to a peripheral 50 HZ magnetic flux which causes the plates to shake down further, following which the space created is filled with more core plates and the top end plate is assembled and pulled down.

3.4.2. Core Frame

The fabricated steel core frame is designed to be as light as possible. The core end plate assembly consists of a thick disc of non-magnetic steel, with separate non-magnetic fingers to support the teeth.

The completed core and the core assembly must be jacked into position inside the casing, where it is supported on feet with resilient mountings, or by flat vertical support plates, either of which provide attenuation of vibration.

3.4.3. Stator Winding

The stator winding must be able to carry the rated current without exceeding specified temperatures and be able to with stand the voltage to earth induced in it. The currents and voltages in the three phases must be exactly the same, but with a 180 deg. displacement for a balanced conditions.

Slots must therefore accommodate six similar winding circuits, differ only in place displacement and 42, 48 or 54-slots are common arrangements. By cooling with water in contract with the conductor, a current density of 8 A/mm square of cross-sectional area can be achieved.

In order to minimise the eddy current circulation because of the leakages fluxes in the teeth the conductor is divided in to strips, which are lightly insulated, arranged in two or for stacks in the bar width. The strips are transposed along the length of the bar by Roebel method, in which each strip occupy every position in the stack for an equal axis length, so that the eddy current voltage are equalised and no eddy currents circulate between strips.

The conductors are made of high conductivity hard drawn copper, each strip or tube has a thin coating of glass fiber insulaiton and is cranked to enable all the strips in a bar to be assembled with the Robel transpositions correctly made. The bar ends are bent using farmers to give the required shape of the end winding. The strips are bound together and the main insulation is applied, a tape of mica powder loaded with synthetic resin, with a glass fiber backing is wound with out breaks along the length of the bar. The straight part of the bar is pressed in a heated mold to cure the resin and obtain design dimensions, while the curved ends are consolidated using heat-shrinkable tape.

3.4.4. Electrical Connections and Terminals

Electrical connections between one conductor bar and the next in series are made differently in different designs. In one, a common electrical and water connector is formed by a copper tube bent into a V-shape, and brazed on to small copper water boxes into which all the bar sub conductors are brazed.

3.4.5. Stator Winding Cooling Components

Water is the best of the commonly available media for cooling the stator winding and imposes only one condition. It must be pure enough to be effectively non-conducting. It is continuously degassed and treated in an ion exchanger with the following target valves being aimed at

Conductivity	:	100 us / m
Dissolved oxygen	:	200 ug / liter max.

Total copper	:	150 ug / liter max
pH value	:	9 max

Water is passed into one or more inlet manifolds, which are copper or stainless steel pipes running circumferential around the core plat. From the manifolds, flexible PTEE houses are connected to all water inlet ports on the stator conductor joints.

3.5 STATOR CASING

The casing contains the stator core and core frame and must resist the load and fault torques. It must also provide a pressure tight enclosure for hydrogen.

Casing are fabricated steel cylinders of upto 25mm thickness, reinforced internally with annular rings and axial members which strengthen the structure and form passages for the flow of hydrogen. Internal spaces are provided with runners to accommodate the hydrogen coolers.

3.6 . INSULATION

The following are the insulation materials that are used in the generator windings.

1. Comprise of organic materials such as cotton, silk paper and certain synthetic films, varnishes and synthetic resins used as binders.
2. Class B insulation. Comprising of magnetic materials such as mica, glass-fiber, asbestos, synthetic films with suitable binders.
3. Class H insulation. This include silicon elastometers as well as mica, glass fiber and asbetos and high temperature binders.
4. Class F insulation. Generally materials used for hot spot temperatures.

In case of windings when voltage exceeds 500 V. special construction is used to control corona this is an electrostalic discharge due to voltage gradient within or at the surface of coils exceeding the dielectric strength of air. In the presence of moisture their discharge produces nitrous acid, which decomposes organic material associated with insulation system. Insulation for high voltage will be applied only when the internal voids are minimised. The outer surface of the slots portion of the high voltage windings are coated with semi conducting medium to lower the voltage gradient between coils and core.

CHAPTER - 4
TURBINE SYSTEMS

4.1 GLANDS AND GLAND SEALING SYSTEMS

4.1.1 Glands

Glands are used on turbine to prevent or reduce the leakage of steam or air between rotating and stationary components which have a pressure difference across them; this applies particularly where the turbine shaft passes through the cylinder. If the cylinder pressure is their than atmospherics pressure there will be a general steam leakage outwards; if the cylinder is below atmospheric pressure there will be a leakage o fair inwards, and some sort of sealing system mµst be used to prevent the air form entering the cylinder and the condenser.

4.1.2 Water –Sealed Glands

Some turbine designs incorporate a shaft gland which depends on a water seal to prevent steam or air leakage. A typical seal arrangement consists of a shaft-mounted impeller with a series of vanes or pockets machined on both faces. The impeller is contained within an annular chamber, and when water is admitted to the chamber, the impeller vanes force the water to rotate at a speed approximately equal to the impeller speed. The seal is relatively inefficient at low speeds and air-sealed auxiliary labyrinth glands must be used, in conjunction with high capacity air pumps, to raise vacuum when starting. Water is usually injected into the seal at approximately half of the full operating speed.

The side clearances between the impeller and seal chamber must be fairly small, and so the use of this seal is expansions are within the effective limits of impeller and seal chamber clearance. When this type of seal is used on a high pressure turbine, the seal cannot absorb the full differential pressure so air-sealed labyrinth glands are used to break the pressure down to a figure which the water seal can handle.

Since a water seal absorbs and generates heat, the water contained in the annular chamber of the water-sealed gland is continuously evaporated; the water losses are made up from a header tank.

4.1.3 Labyrinth Glands

In modern turbines the labyrinth gland are used because it can withstand high pressruns and temperatures and yet requires little maintenance.

The labyrinth gland provides a series of very fine annular clearances, in the gap between the cylinder wall and the shaft. The steam is throttled through this gap and its pressure reduced step by step. In expanding through each clearance, the steam develops kinetic energy at the expense of its pressure energy; ideally, the kinetic energy is converted by turbulence into heat with no recovery of pressure energy. In this way, the at successive restrictions. By keeping the clearance area sufficiently small, the quantity of energy lost may be kept low and as increase in turbine output occur the gland leakage loss becomes proportionately less.

To reduce the clearance area, glands are made with a diameter as small as possible, and clearances as fine as possible. The diameter is limited by considerations of shaft strength and radial clearances by the clearances within the bearing, and by the possibility of shaft distortion.

Glands must allow for axial expansion of the shaft and casing to take place without causing a rub. On the other hand, if a rub does take place because of shaft vibration, it is desirable that the heat generated is minimized to prevent serious frictional heating of the shaft and possible distortion. A typical modern gland comprises stationary fins on spring-loaded sectors, while the shaft is either smooth or castellated. If a rub should occur, the sectors receive the generated heat and can be replaced readily if they are damaged.

4.2 GLAND SEALING SYSTEM (210 MW LMW/BHEL TURBINE)

The 210 MW turbine has got 6 sets of gland seal one for each side of HP and LP. Each set of gland seal consists of no. of sectionalized gland. HP turbine front gland is sectionalized in to five section. HP rear and IP front into four sections, IP rear into three sections and LP glands into two sections. Each glands sealing consists of number of sealing rings divided into segment, each segment is backed by two flat springs. The no. of sealing rings depended on pressure against which it is working. The sealing rings are housed in grooves machined in gland bodies which in turn are housed in the turbine casing or bolted to the casing at ends.

The glands are sealed by steam. Penultimate section of each gland is supplied with steam from gland steam header maintained at a pressure of 1.03 Kg/cm^2 abs to 1.05 Kg/cm^2 abs and between 130^0C to 150^0C. The header receives steam from deaerator steam space through a pressure control valve. Provision is also there to supply the steam form TAS (Turbine Aux steam) system during non availability of deaerator steam.

Steam supplied to last but one seal of each gland at a pressure of 0.10 to 0.20 Kg/cm^2 and temperature 130^0C to 150^0C from sealing steam header. Leak of steam form the turbine is tapped off from different section in the gland seal and is either cooled in the gland steam cooler or fed to lower pressure stages of turbine. The leakage form the last stage i.e. air side leakage is cooled in gland cooler No.1. The cooler is maintained under vacuum with help of a special steam ejector provided inside the gland cooler No.1.

Steam form fourth sealing chamber (from air side) of HP front, real and IP front is connected with turbine 4^{th} extraction before the NRV. Steam from the third chamber of HP front, real and IP front rear is connected with the Gland steam cooler No.2 for regenerative feed heating cycle. The third gland chamber and gland cooler No.2 are always maintained at condenser vacuum as the condensate drain side of gland cooler No.2 is connected with the condenser. The leak off steam form the first chamber of the HP front gland is connected to HP turbine last stage. Steam/air mixture form the spindle seals of ESVs, IVs and control values of HP and IP is exhausted into the gland steam cooler No.1.

In addition to the above mentioned gland steam supply system, an another source of supplying gland steam from the live steam is there.

The gland cooling system arrangement is shown in the following fig 4.1

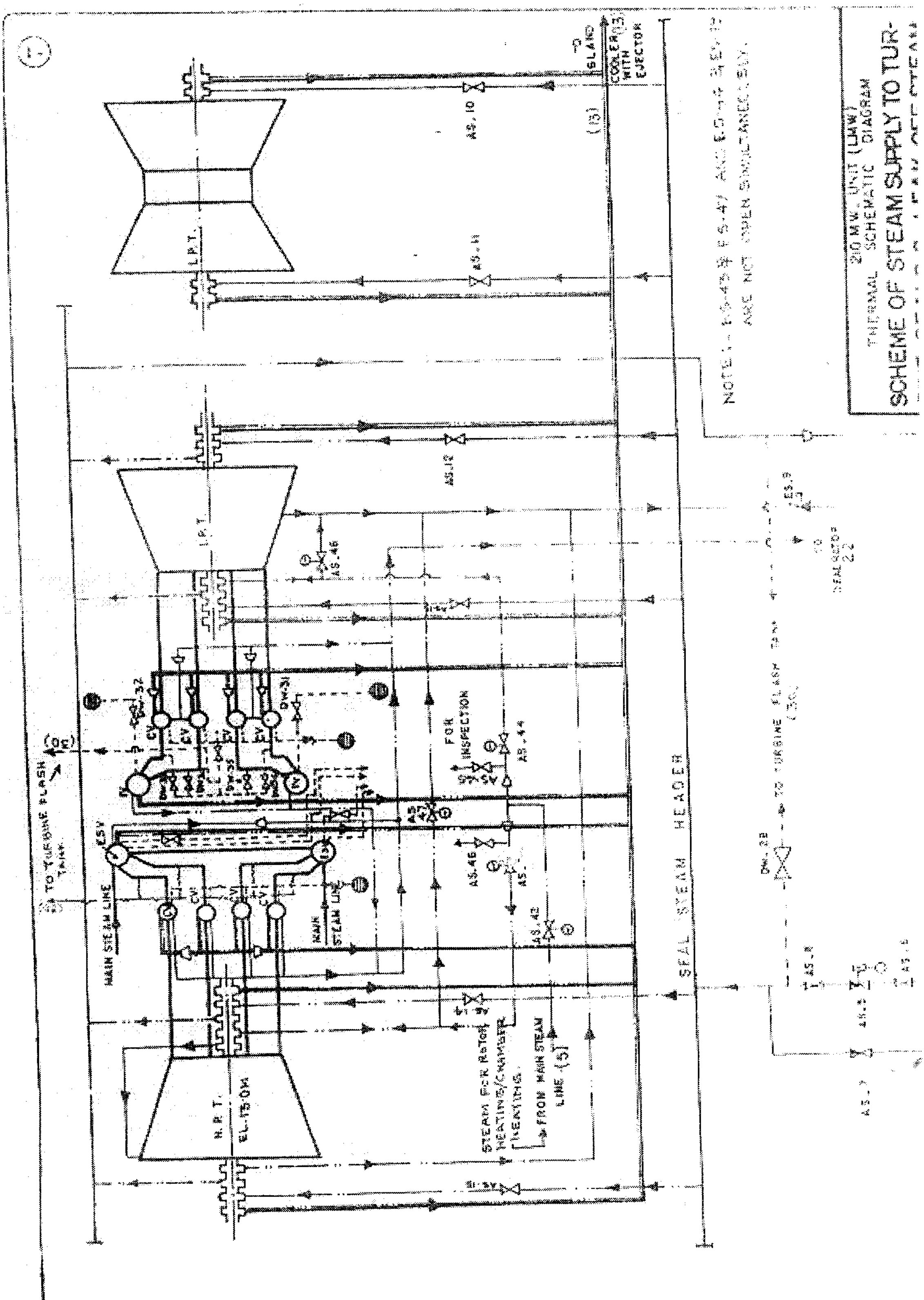


FIGURE 4.1 GLAND SEALING SYSTEM

All the gland seals of turbine are normally fed with steam form deaerator at controlled pressure. Following a turbine trip-out, this gland steam shall be sucked into the cylinders which would be under vacuum at that condition. Since the temperature of gland steam fed from deaerator is much lower than expansion of HPT and IPT are likely to increase at a faster rate under such conditions. In view of the above a provision has been made to inject main steam to the front gland seals of HPT and IPT. This system is called Rotor heating system.

4.3 CONDENSATE SYSTEM

A typical condensate system consists of the following

i) Condenser (including hot-well)

ii) Condensate pumps

iii) Air extraction system

iv) Gland coolers and L.P. heaters

v) Deaerator

4.3.1 Condenser

The functions of condenser are

i) To provide lowest economic heat rejection temperature for the steam. Thus saving on steam on steam required per unit of electricity.

ii) To convert exhaust steam to water for reuse thus saving on feed water requirement.

iii) Deaeration of make-up water introduced in the condenser.

iv) To form convenient point for introducing make up water.

4.3.2 Description of Condenser for 210 MW (BHEL) Turbines

The condenser group consists of two condensers, each connected with exhaust part of low pressure casing. These two condensers have been interconnected by a by-pass branch pipe. The condenser has been designed to create vacuum at the exhaust of steam turbine and to provide pure condensate for re-using as feed water for the boilers. The tube layout of condenser has been arranged to ensure efficiency heat transfer from steam to cooling water passing through the tubes, and at the same time the resistance to flow of steam has been reduced to the barest minimum.

3x50% capacity condensate pumping sets are installed for pumping the condensate from condenser to the deaerator through low pressure heaters. Tow pump are form normal operation and one works as stand by pump.

4.3.3 Constructional Feature

Each condenser has been sub-divided into upper and lower parts as shown in the fig 4.2. Front water box, shell and rear water box constitute the lower part. Tow end tube plates and six support plates are located inside the lower body of the condenser.

Front water boxes have been divided into two parts to make the condenser two pass designs. End covers of water boxes are tubes. Manholes have been provided for routine maintenance and visual inspection along with venting and draining arrangement for individual water boxes. Condenser tubes are secured to the end tube plates by expanding and flanging of tube ends which provides very good sealing arrangement against penetration of circulating water into the steam space. The tubes have been so arranged that there is equal distribution of steam on the tube nest with minimum resistance to steam flow. Non-condensable gases are continuously sucked with the help of steam ejectors.

With a view to allow relative expansion between tubes and the body of the lower part, lens type compensator has been provide din the body itself at the rear water box end This arrangements prevents deformation of the body and damage to connections between tubes and end plates.

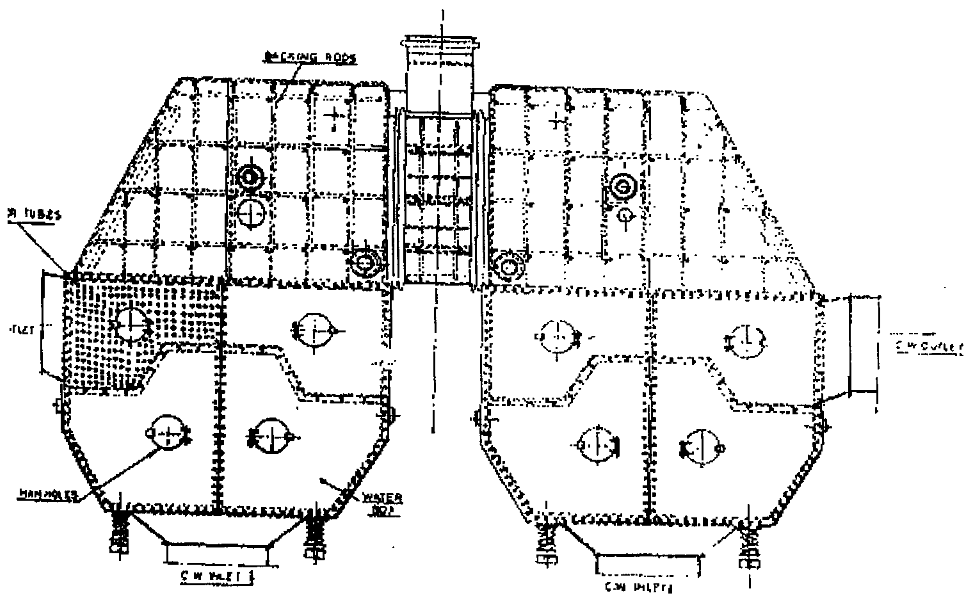


FIGURE 4.2 CONDENSER

Upper part of condenser has been designed to allow smooth flow of steam over tube nest. It consists of mild steel flat walls, strengthened form inside by gratings of longitudinal and transverse rods and form outside by channels. These rigid bars help the condenser to retain its shape against atmospheric pressure.

Two sections of low pressure heater No.1 have also been located inside the upper parts of condenser.

In order to allow expansion along the height, the condenser is supported on springs specially designed to take up load.

The weight of the condenser and its tubes is taken by the springs and through them by the condenser foundation. The weight of circulating water and the condensate along with the thrust of springs during expansion is transferred to turbine foundation.

Special care has been taken for removal of condensate formed as a result of condensation of steam. Baffle plates have been provided to guide the steam flow on the tube nest and for collecting the condensate tricking from upper rows of tubes and directing it towards the intermediate support plates for flowing down in narrow layers, leaving the passage free from steam flow.

A steam throw off device has been incorporated in each condenser for dumping the steam into the condenser during start up and sudden load throw off from the set.

4.4 CONDENSATE EXTRACTION PUMPS

Condensate extraction pumps are normally multistage, vertical, centrifugal pumps. They are generally required to operate on minimum net positive suction head (NPSH). The condensate pumps operate on few inches of suction submergence. It is shown in the fig.4.3

A vent line connects the hotwell, from where the condensate pups take suction with the condenser. This equalizes the vapour pressure of condenser and hotwell.

No. of stages in the pump is determined by the discharge pressure required for the condensate cycle.

In 210 MW unit, three condensate pumps, each having 50% capacity, are provided for pumping the condensate to deaerator. Condensate water is also used for:

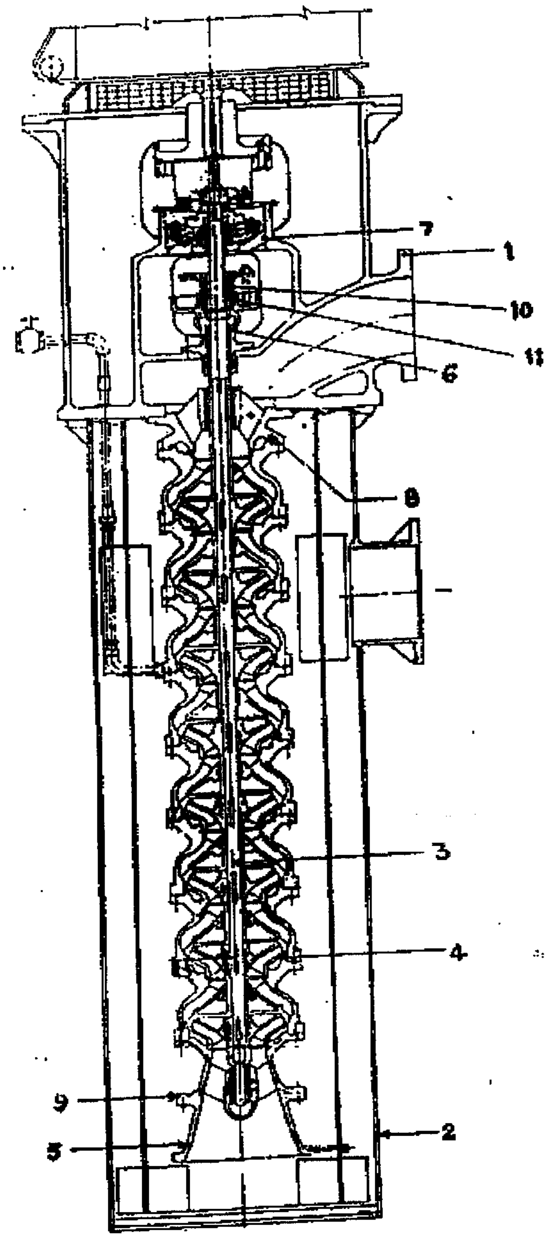
i) Sealing of glands of valves operating under vacuum.

ii) Temperature control of L.P. bypass steam.

iii) Filing syphons of main ejectors and 15 meter siphon of drain expander.

iv) Actuating the forced closing non-return valves of turbine steam extraction lines.

v) For cooling steam dumped through steam throw off device.



1. Discharge Head
2. Outer Shell
3. Impeller Shaft
4. Impeller First Stage
5. Bell Mouth
6. Stuffing Box Housing
7. Thrust Bearing Housing
8. Upper Pump Bearing
9. Lower Pump Bearing
10. gland in Two Halves
11. St. Box Packing

FIGURE 4.3 CONDENSATE EXTRACTION PUMPS

4.4.1 Major Specification Of A Typical Bhrc 28 Type Condensate Extraction Pump (For 210 Mw)

Pump:

3 Nos. per unit

Multistage, vertical turbine centrifugal pump.

Low specific speed, medium head,

Medium capacity

Discharge – 281 T/hr

Manometric =- 210 Metres

Head

NPSH – 3.5 MTRS

r.p.m. – 1489

No. of stages – 8

No. of stage – 8

hp – 256

Motor

Power - 220 kW (300 hr), Voltage – 6.6 kV

4.5 AIR EXTRACTION SYSTEM

Air extraction system is needed to extract air and other non condensable gases form the condenser for maintaining vacuum.

Amount of air to be extracted form condenser during start up is quite large and the extraction should be down as rapidly as possible so as to allow the turbine to be started.

Under normal operating conditions quantity of air to be extracted is lower. Ti consists of air leakage into the condenser via flanges and glands and also of very little non condensable gases present in steam.

To guard against excessive water vapour extraction along with air, the space beneath the air extraction baffles has been provided with its own cooling tubes in order to condense as much water vapour as possible and thus preventing its removal form condenser.

4.5.1 Air Ejectors

The operating medium of the air ejector can be either high pressure gas or liquid. In thermal power stations steam of low parameter (Approx. 4.5 Kg/cm^2, 250^0C) is used for the air ejector. The operation principle is simple – steam is passed through a nozzle and the pressure energy converted into velocity energy. High velocity fluid aspirates air and other non condensable gases form the condenser and moves into a diffuser which re-converts the velocity energy into pressure energy. The pressurized mixture of steam and air is exhausted, either directly to atmosphere or through coolers to recover the steam in the form of condensate.

4.5.2 Starting Ejector

Starting ejector is recommended to be used of reaccelerating initial pulling of vacuum. During this period starting ejector operates in parallel with main ejector. When the vacuum in the condenser reaches 500-600 mm of Hg column, the starting ejector is switched off.

Figure shows the construction of a starting ejector. It may be noted that steam along with the mixture of air and other gases is exhausted to the atmosphere. Generally starting ejector is single stage and has high steam consumption.

The starting Ejector is shown in the following fig 4.4

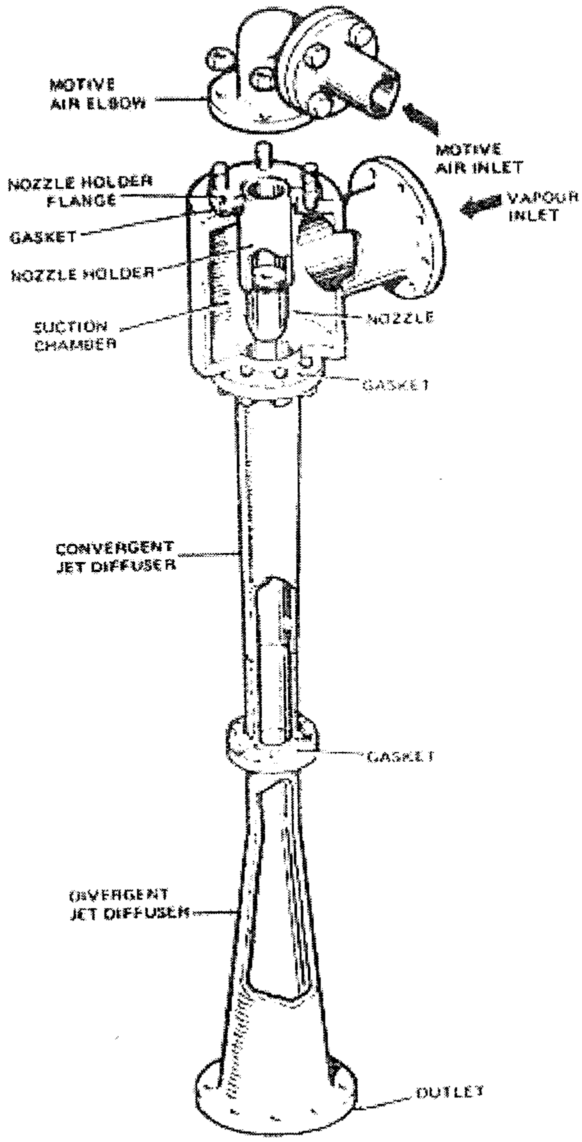


FIGURE 4.4 STARTING EJECTOR

4.5.3 Main Ejector

The main ejector with a standby unit is usually provided for normal operation. The main ejector is a multi stage type, the number of stages depends on the cooling water condition. Steam at suitable pressure is passed through a converging-diverging nozzle, and the pressure energy of steam is converted into velocity energy. This high velocity steam jet entrains air and incondensable gases and then enters a diffuser where velocity energy is converted back into pressure energy. The steam/air mixture is then cooled in the first stage shell by condensate. Steam is thus condensed, heat in the operating system is partly recovered, and the steam/air mixture volume is reduced, allowing the second stage nozzle and its steam consumption to the reduced. The second stage cooler can be followed buy the third stage nozzle, and its after cooler (as done in BHEL 210 MW unit). Drains are usually returned to the condenser via suitable loop seals; condensate as a cooling medium is taken form the extraction pump discharge, with a recirculation arrangement to avoid overheating of the eject at low loads. Fig 4.5 shows the arrangement of typical tow-state main ejector.

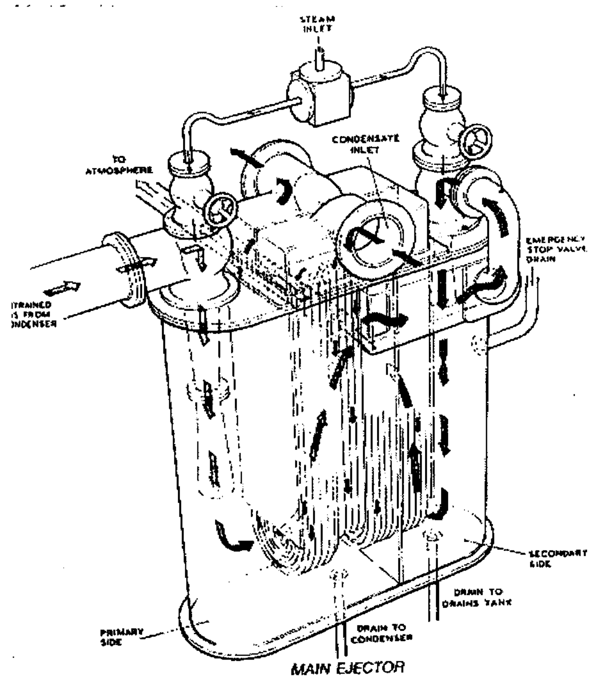


FIGURE 4.5 MAIN EJECTOR

An air measuring device for measurement of air discharge form condenser may be fitted at the air exit of the ejectors. It measures dry discharge while the condenser and ejectors are in operation.

4.6 BOILER FEED PUMP

Boiler feed pump (BFP) is a multistage pump provided for pumping feed water to economizer. Generally three pumps each of 50% of total capacity are provided. For rated capacity tow pumps will be working in parallel and the third will be in reserve.

4.6.1 Description Of Feed Pump (Bhel – 200 Khi, Provided For 200/210 Mw Unit)

Feed pump consists of the following major parts:

1. Pump Barrel
2. Rotor
3. Stator
4. Mechanical Seal
5. Balancing Device

Pump Barrel

The barrel is essentially a cylinder which houses both the stator and rotor. The suction side of the barrel and the space in the high pressure cover behind the balancing device are closed by the low pressure covers along with the stuffing box casings. The brackets of the radial bearing of the suction side and the bracket of the radial and thrust bearings of the discharge side are fixed to the low pressure covers. The entire pump is mounted on a foundation frame. As the pump handles hot water, sometimes, arrangements are made for cooling the foundation frame to prevent unequal expansion of the frame.

Rotor

The rotor of boiler feed pump consists of the shaft, impellers, distance bushes, balancing disc, supporting rings etc. the axial thrust of the rotor is taken up by the balancing disc. Which is keyed to the shaft in between the tow parts supporting rings which are mounted in the grooves in the shaft. The rotor is supported on two part bearing shells. The bearing brackets are connected to the low pressure cover. The cut section of Boiler Feed Pump is shown in Fig.4.6.

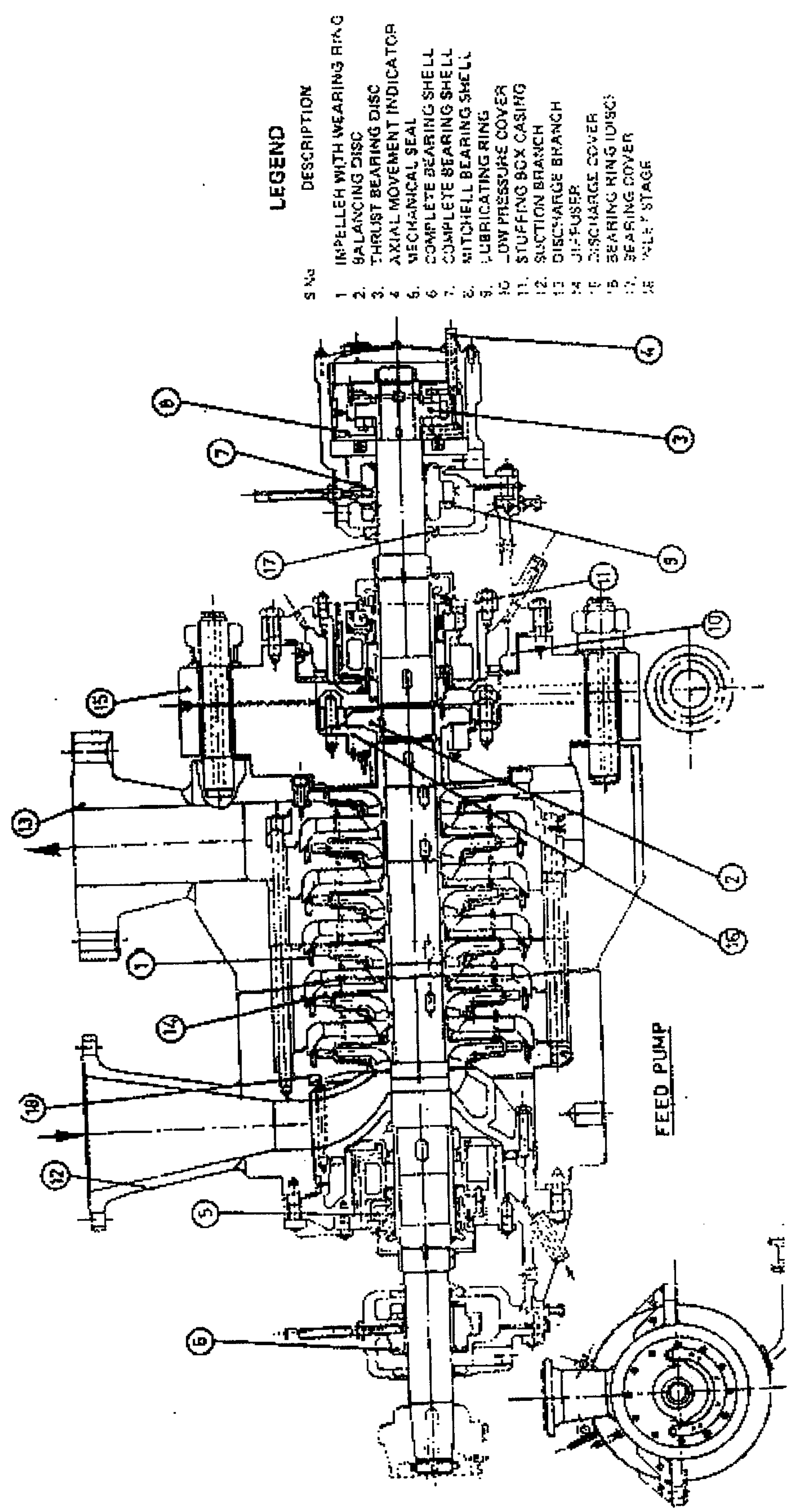


FIGURE 4.6 BOILER FEED PUMP

Stator

The stator consists of stage bodies. The diffusers with the diffusing wheels and guide wheels are assembled to the stage bodies. The end diffuser is connected to the outlet stage outside the stage body. Stage bodies are fitted with wearing rings at the place where it is likely to come into contact with the wearing rings of impellers, and the wearings rings are secured to the stage bodies with the help of screws.

Mechanical Seal

Sealing of the pump is achieved by a specially designed mechanical seal. It minimizes the loss of the feed water increases. With the use of the mechanical seal, the cooling is carried out by the circulation of water between the stuffing box space and the cooler. The feed water is circulated in the cooling circuit through the cooler and back by means of a pumping ring. The cooler are so designed that water temperature in the stuffing box remains below 80^0C.

Balancing Device

As in other stage pumps, all the six impellers are arranged on the shaft with inlets in the same direction. This causes a thrust of about 34 tones in the direction of suction end of the pump while running. This axial thrust is taken up by the balancing device. About 10% of feed water which is not calculated to the guaranteed delivery capacity is taken off from the space behind the last impeller for operation of the automatic balancing device. The balance disc is fixed to the shaft and rotates between a renewable seating and the balance disc cover. The thrust generated by the impellers tends to force the disc against its seating, but the high pressure water, bled off the delivery stage of the pump, flows along an annular space between the hub of the disc and the bush, which is an integral part of the balance disc seating, to a pressure chamber.

The pressure in the chamber builds up until it exerts sufficient pressure on the balance disc to overcome the end thrust of the impeller. Water hen escapes between the face of the disc and its seating. The balance disc thus turns on a film of water and does to come into metallic contact with the seating. Water leakage across this disc is called balance water and is returned to the deaerator.

A thrust kings berry bearing takes over the function of the balancing device when feed pump is started. The kings beery shell is forced against the direction of action of balancing disc on the disc by means of springs located in the kings beery bearing. By action of springs, an axial gap of about 1.0 mm is formed between the contract surfaces of the bearing disc. The total pull of springs is equal to 500 kg. With the staring of the pump the axial thrust increases gradually and thrust kings berry bearing is in action until the time when the magnitude of the axial thrust overcomes the pressure of the springs mounted in Mitchell bearing, the rotor will move to the suction side and balancing disc comes into contact with bearing disc, reducing the axial gap an due to the increased pressure on the balancing disc, the rotor move to the middle position creating the gap between the balancing disc and the bearing ring.

Even under worst condition when the rotor moves to the suction side and the balancing disc is likely to come into contact with the bearing ring before the necessary pressure being built up on the balancing disc to overcome the axial thrust, a certain amount of water flows through the axial gap between the balancing disc and the bearing ring and there is no danger of balancing device getting seized.

It is evident that behind the balancing disc the pressure must not rise, otherwise the hydraulic equilibrium will be broken and therefore equalizing piping must have a sufficient flow capacity. For safe operating, the pressure in the equalizing piping should be 0.5 to 2 atm, higher than the intake suction branch pressure. When the pressure in the balancing space rises by 5 atm above suction pressure it is necessary to trop the pump in order to find out the case of defect and to ratify its.

4.6.2 Working Of Boiler Feed Pump

The water with the given operating temperature should flow to the pump under a certain minimum pressure (NPSH), water passes through the suction branch into the intake spiral and form here is directed to the first impeller. After leaving through the impeller it passes through the distributing passages of the diffuser where it get certain pressure rise and flows over to guide banes to the inlet of the next impeller.

This process repeats form one stage to the other till it passes through the last impeller and the end diffusers. Thus the feed water arriving into the discharge space develops the necessary operating pressure. A small part of feed water i.e. about 10% is taken off from the space behind the last impeller for the operation of the automatic balancing device to balance the hydraulic axial thrust of the pump rotor.

4.6.3 Typical Specifications Of Boilers Feed Pump (200 Khi Type)

No. of stages	6
Suction pressure	12. 3 ata
Quantity of water for minimum take off	100 Tones/hr.
Discharge capacity/head	430 T/hr./1830 MWC
Quantity of water for warming up	8 Tones/hr.
Feed water temperature	164.2^0C
Consumption of cooling water	280 LPM
Speed	4320 rpm
Lubrication	Forced
Stuffing box	Mech. seal
Net weight of pump	5850 kg.
Axial thrust at designed speed.	34 Tonnes
Motor	
Output	4000 kw
Rated voltage	6.6 kV
Current	421 Amps.
Speed	1483 rpm
Frequency/Power factor	50 c/s/0.914

4.7 RECIRCULATION SYSTEM

To maintain a reasonable efficiency in the pump, running clearances between stationary and rotating parts must be fairly narrow. Liquid flow through these clearances acts as a lubrication to prevent seizure. The power input to the pump is partly converted into hydraulic energy due to the increase in pressure of the liquid. The remaining energy is wasted in the form of frication, eddies mechanical losses. This power loss cases slight increase in the liquid temperature while the liquid passes form suction to discharge. This temperature rise is maximum at zero discharge and the water soon flashes into steam. Flashing breaks down the thin film of lubricating water between the pars and this usually causes seizure. The trouble occurs so quickly that stationary parts cannot expand as rapidly as the rotating parts, because they will be heated more slowly, being of greater mass and also exposed to atmosphere. Greater expansion of rotating arts will refuse the normal running clearance and aggravate the conditions.

It is, therefore, imperative that sufficient water must be kept moving through the pump to prevent its temperature from reaching the flash point in the pump when the regulator closes the main discharge line due to low load or less water requirements in the drum or when the pump is just started. To ensure this an automatic leak off system is provided between the pump discharge and the deaerator to establish a minimum flow through the pump. A solenoid operated diaphragm valve or a motorized valve is installed in the leak off line which opens when the pump runs at a lower capacity.

The recirculation valve (of BHEL 210 MW unit BFpp) opens when the flow at pump suction is below 100 T/hr & closes when it increases to 220 T/hr. the flow through recirculation line is 125 T/hr.

4.8 REGENERATIVE FEED HEATING SYSTEM

4. 8.1 Economics of Feed Heating

If steam is bled from a turbine and is made to give up its latent and any superheat it may possess, to a heater, this system is called regenerative, because the fluid (steam) rives up heat, which would be otherwise wasted, to the fluid whilst in another state (water) to raise its temperature. The highest theoretical temperature to which the feed water may be raised in the heater is the saturation temperature of the bled steam. There is an optimum point at which the steam is bled from the turbine once a feed temperature is selected. A tappling point near the stop valve produces no gain in efficiency as practically live steam is used for heating. An intermediate point, if carefully chosen, gives maximum feed temperature rise with minimum loss of mechanical power at the turbine. The steam, having given up a proportion of its work to the turbine, then gives up all its latent heat which would otherwise be lost to the condenser C.W. The heat gained in this way outweighs the loss of mechanical power and a gain in efficiency follows. Other advantages of this cycle are that loss C.W. is required with a decrease in pumping power, a smaller condenser can be used and the turbine exhaust annulus is smaller.

The thermal gain resulting from feed water heating can be (illustrated by considering an example with approximate figures as follows (Single Feed Heater)). The economics of fed heating is given in table 4.1.

TABLE 4.1 ECONOMICS OF FEED HEATING

	Without Feed Heating	With Feed Heating
Turbine steam consumption	4.5 kg/kWh	4.5x1.05= 4.725 kg/kWh
Boiler feed temperature	31^0C	93^0C
Total heat in steam	3140x4.5=14130 kJ/kWh	3140x.75=14836.5 kJ/kWh
Heat already in water	4.5x4.19x31 = 584.5 kJ/kWh	4.725x4.19x93=1840 kJ/kWh
Heat used by turbine = heat in mass of steam- heat to raise water	(4.5x3140)-584.5 =14130-584.5 (13545 kJ/kWh)	(4.725x3140)-1840 =14836-1841 =12995 kJ/kWh

The difference between these values is 4.23% which can result in a considerable yearly saving in fuel consumption.

4.8.2 Types Of Feed Water Heaters

A feed water is simply a heat exchanger which is arranged so that the water leaving a condenser is pre-heated before it is fed to a boiler. The feed heater is supplied by steam which has already performed some useful work. This steam which is taken from suitable stages along a turbine, transfers its latent heat to the boiler feed water and accordingly increases the water temperature.

It is now universal practice to use feed heaters to heat the feed water from the temperature at which it leaves a condenser to a temperature approaching the saturation temperature of the boiler steam pressure.

When a feed heater is in operation, it requires no regulation because the bled steam consumption responds automatically to the temperature and quantity of feed water passing through the heater.

Low pressure feed heaters are positioned after an extraction pump, while high pressure feed heaters are positioned after a boiler feed pump and, therefore, have to be constructed to withstand the full discharge pressure of boiler feed nump.

Two types of feed heaters are used; the surface type, in which the feed water is passed through tubes, with the bled steam surrounding them; and the direct contact type, in which the steam and water mix together.

The Surface – Feed Heater

A surface type heater consists basically of three parts

a) A shell with a steam connection
b) A nest of tubes
c) A water box, with water connections on the inlet and outlet headers.

A low pressure surface heater may consist of a cylindrical body fabricated from mild steel and sealed at its upper end by a cast steel water box, which houses a nest of solid brass U-tubes. The ends of the tubes are expanded into a mild steel tube plate trapped between flange on the body and a corresponding flange on the water box. Baffles are provided to ensure that the steam is directed across the tubes. The upper section of the quadrant of the tube nest, which carries the condensate in its last pass through the heater, is totally enclosed by vertical baffles, so forming a flashed-steam drain cooler section of the heater.

A high pressure surface heater consists of a mild steel cylindrical and some secured together to form a vessel, which houses a tube nest of carbon steel tubes suspended from a condensate inlet and outlet header. LP heater arrangement is shown in Fig.4.7.

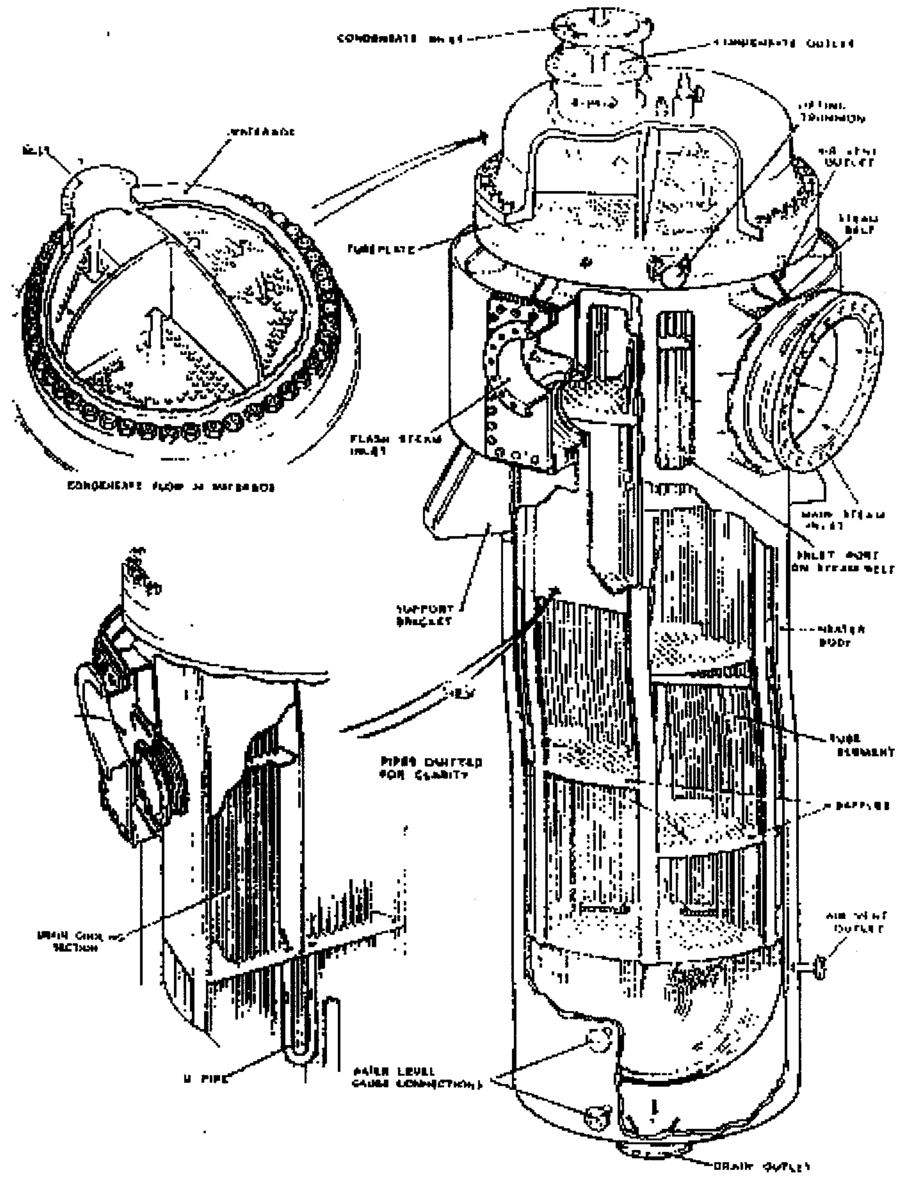


FIGURE 4.7 LP HEATER

A typical modern high-pressure heater the integral desuper heating and drain cooling zones in addition to condensing surfaces in the one steel. Temperature/heat transferred diagram for this heater. The outlet terminal difference is -0.56^{0}C (-1^{0}F) and the drain cooling terminal difference is 5.56^{0}C (10^{0}F)

4.8.3 Heat Transfer In Feed Water Heaters

There are three heat transfer zones in a typical H.P. surface feed heater. The following figure 4.8.shows the HP heater.

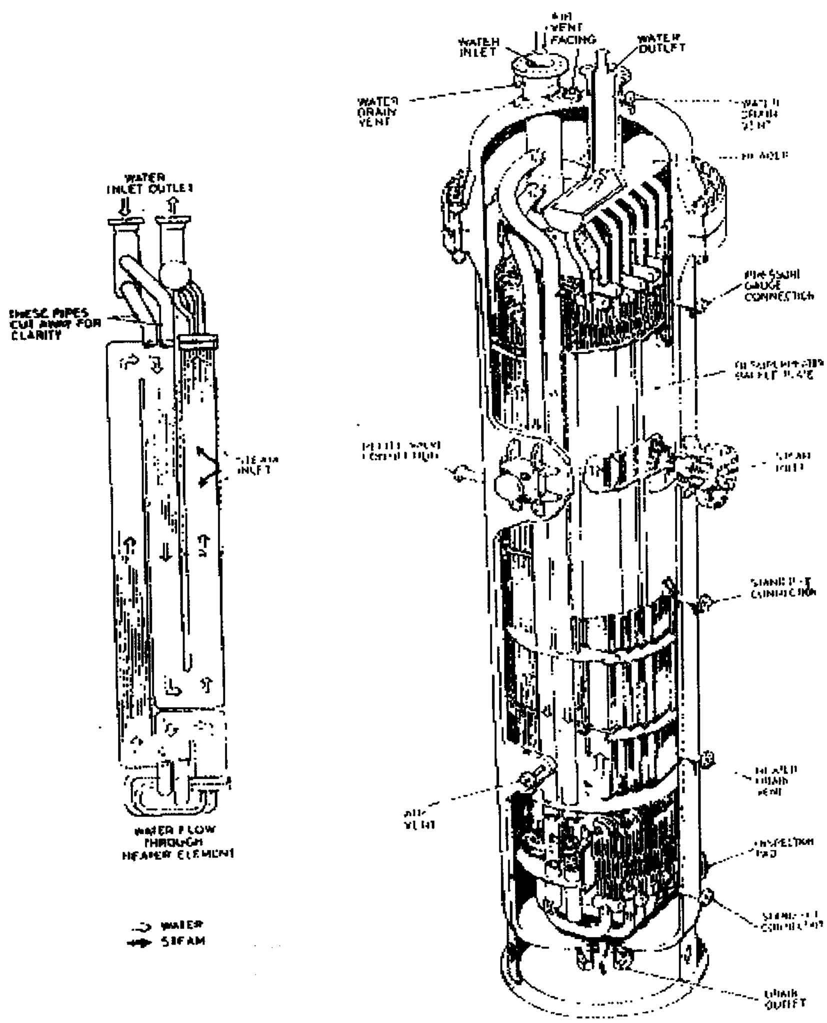


FIGURE 4.8 H.P HEATER

a) Desuperheating Zone

This zone is the last feed heater zone through which the feed-water passes before leaving the heater. When the feed water enters the zone it has been heated in the condensing zone to within a few degress of the saturation temperature corresponding to the bled steam pressure at the entry to the feed heater. Although the desuperheating zone adds only a small percentage of the total heat transferred in the heater, this small rise in feed-water temperature is very valuable in terms thermal economy. The heat in the zone is transferred by convection- the superheater steam can be considered to behave as a normal gas.

The steam normally leaves the desuperheating zone with a residual superheat of about 27.8^{0}C. The temperature gradient between the bulk steam and feed water temperature is shown in the top right hand inset of. The tube wall temperature is above the saturation temperature of the bled steam and no condensation takes places on the tube wall under normal conditions. There is, therefore, no problem of condensed steam forming droplets and being carried into the condensing zone by the relatively high velocity steam which could cause impingement attack at the exist from the zone. Heat transfer diagram is given in Fig.4.9.

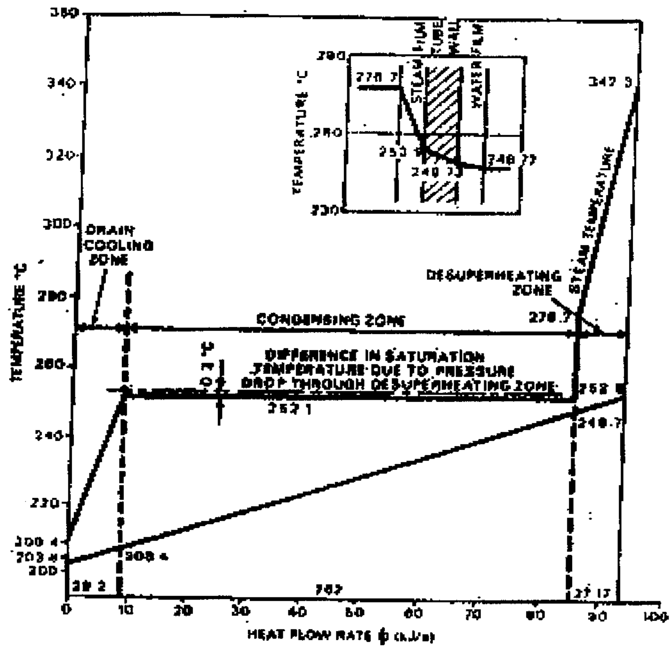


FIGURE 4.9 HEAT TRANSFER DIAGRAM FOR HP HEATER

b) Condensing Zone

In the condensing zone the overall coefficient of heat transfer is high. The main problem in larger heaters is to obtain good steam distribution to the condensing surface with the minimum pressure loss. Under normal load running conditions the ingress of non-condensable gases is unlikely. However, continuous venting is necessary to help the steam distribution and to clear any non condensable gases which are likely to accumulate after shut down during two shift operation. The disphragm plates serve a dual purpose, not only do they support the tube nest, but they also prevent the accumulation from running down the outsiders of the tubes and forming a thick film with a consequent reduction in heat transfer.

c) Drain Cooling Zone

It is possible to obtain a fairly high heat transfer rate in the drain cooling zone provided the drains are not reheated by heat transferred through the shrouds around the drain cooler convection zone. If there were no desuperheating by a steamy atmosphere which would tend to condense on the shrouds and reheat the drain. In the heater shown in the desuperheating zone is larger than the drain cooling zone and it is possible to have the heater shell flooded with condensate to above the drain cooler inlet. This condensate forms a stagnant pool around the drain cooler shrouds causing them to follow the temperature gradient in the drain cooler. Under this condition the problem of drain reheat is eliminated provided the shell drain valves are kept closed.

The regenerative system of the turbine consists of four low pressure (LP) heater, two gland coolers, one deaerator and three high pressure (HP) heaters. The condensate is drawn by condensate pumps form the hot well of condenser and is pumped to the deaerator through gland cooler and low pressure heaters where it is progressively heated up by steam extracted form seals and bled points of the turbine. The drain of condensed steam on LP heaters No.2, 3 and 4 flows in cascade and is ultimately pumped into the main condensate line after heater No.2 or flows to condenser. The feed water after being deaerated in the deaerator is drawn by boiler feed pump and pumped to bonier through high pressure heaters where it is heated up by the bled steam form the turbine. The drain of condensed steal of HP heaters flows in cascade and under normal load conditions flow to the deaerator.

4.8.4 Low Pressure Heater No.1

The heater is of horizontal surface type consisting of two halves, each half has been located inside the upper part of each condenser. The tow halves have been installed in parallel. The steam to both is supplied form the same extraction point.

The housing for the heater is fabricated from M.S. plates with suitable steam inlet and drain connections. The tube plate is of mild steel and is secured to the water box and housing by means of studs and nuts. 'U' shaped tubes have been used to ensure independent expansion of tubes and the shell. They are of solid drawn admiralty brass, 19mm external dia, 1mm and 0.75 mm thick and are the expanded b rolling into the tube plate at facilitate drawal for tube replacement, and maintenance. Partitions mild steel plates have been provided for supporting the tubes at intermediate points and effective distribution of heat load in all the zones of to heater.

The water box is of mild steel with suitable water inlet and outlet branches. Ti s of rectangular shapes and has been provided with suitable air vent and drain connections.

4.8.5 Low Pressure Heater Nos. 2,3,&4

a) Construction

These heaters-identical in contraction are of vertical surface type and are designed for the steam to pass over the stubs and the condensate to flow through them. Following are main elements of these heaters.

i) Shell
ii) Tube system
iii) Removable water box.

Shell is a cylindrical construction with dished end welded at. Bottom and having a flange at the upper end for assembly of tube system and water box. The shell is provided with suitable steam inlet and drain connections along with other nozzle connections to accommodate various fittings. M.S. baffles are provided to ensure effective distribution of steam in the condensing zone of the heater.

Tube system consists of U-shaped admiralty brass tube, 16mm external dia, 1 mm thick and are expended by rolling into tube plate at both the ends. Tube system has been provided with rollers to facilitate drawal for tube replacement. Tube plate is of mild steel and is secured to the water box and shell flange by means of studs and nuts.

Water box consists of thick walled cylindrical shell having a flange at the lower end and a dished end welded at top. It ahs been provides with suitable water inlet and outlet branches. Partitions have been provided in the water box to make to four path design.

b) Working Principle

The main condensate flows through the tubes in four paths before leaving heater. The heating steam enters the shell through a pipe and flow over the U-shaped tube nest. The partition walls installed in the tube systems ensures zigzag flow of steam over tube nest. Condensate of heating steam referred as drain, trickles down the tubes and it taken out form the lower portion of the shell by automatic level control valve installed on the drain line.

4.8.6 Gland Steam Cooler No.1

It cools the air-steam mixture sucked form turbine end seals. It is of vertical type and has tow sections. An ejectors mounted on the cooler, maintains constant vacuum in the first section. Ti also sucks the remaining air steam mixture form 1st section to second, where air is let off and steam condensed. A part of main condensate, after main ejector, flows through the cooler tubes consisting of U-shaped brass tubes rolled in steel tube plate. Drain form cooler is led to condenser.

Gland cooler no. 1

Gland cooler has been designed to condensate the leak-off steam from intermediate chambers of end sealing of HP &IP turbines.

The construction of this cooler is identical with low pressure heaters no.2,3,&4.

The main condensate flows through the tubes in four paths before leaving the cooler. The leak off steam enters the shell through a pipe and flow over the tube nest. The participation walls installed in the tube system lead to zigzag flow of steam over the tube nest. Condensate of leak off steam referred as drain trickles down the tubes is taken out form the lower portion of the shell by automatic level control valve, installed on the drain line.

4.9 DEAERATOR

4.9.1 Functions

The pressure of certain gases like Oxygen, carbon dioxide and ammonia, dissolved in water is harmful because of their corrosive attack on metals, particularly at elevated, temperatures. Thus in modern high pressure boiler, to prevent internal corrosion, the feed water should be free, as far as practicable, of all dissolved gases, especially oxygen. This is achieved by embodying into the freed system a deaerating unit. Apart from this, a deaerator also serves the followings functions:

1) Heating incoming feed water
2) To act as a reservior to provide a sudden or instantaneous demand.

4.9.2. Principal of Deaeration

a) The solubility of any gas dissolved in a liquid is directly proportional to the partial pressure of the gas. This holds within close limits for any gas which does not react chemically with the solvent.
b) Solubility of gases decrease with increase in solution temperature or decrease in pressure.

4.9.3 210 MW LMW Unit Deaerator

A constant pressure deaerator, pegged at 7 kg/cm^2 (abs) is provided in turbine regenerative cycle to provide properly deaerated feed water for boiler, limiting gases(mainly oxygen) to 0.005 cc/Litre. It is a direct contact type heater combined with feed storage tank of adequate capacity. The hating steam is normally supplied form turbine tractions but during starting and low load operation the steam is supplied form auxiliary source.

The deaerator comprises of two chambers

1) Deaerating column
2) Feed storage tank

Deaerating column is a spray cum tray type cylindrical vessel of horizontal construction with dishes ends welded to it. The tray stack is designed to ensure maximum contact time as well as optimum scrubbing of condensate to achieve efficient deaeration. The deaerating column is mounted on the feed storage tank which in turn is supported on rollers at the tow ends and a fixed support at the centre. The feed storage tank is fabricated from boiler quality steel plates. Manholes are provides on deaeration column as well as on feed storage tank for inspection and maintenance.

The feed is admitted at the top of the deaerating column and flows downwards through the spray valves and trays. The trays are designed to expose to the maximum water surface for efficiency scrubbing to effect the liberation of the associated gases. Steam enters form the underneath of the tray and flows in counter direction of condensate. While flowing upwards through the trays, scrubbing and heating is done. Thus the liberated gases move upwards along with the steam. Steam gets condensed above the trays and in turn heat the condensate. Liberated gases escape to atmosphere form the orifice opening meant for it. This opening is provided with a number of deflectors to minimize the loss of steam.

In some deaerator designs, a vent condenser is also located above the deaerator. A portion of feed water is first passed through the vent condenser before it enters the deaerator. This water is heated by remaining steam after a steam has passed trough the deaerator. Thus only gases escape to atmosphere.

4.9.4 Location Of Deaerator

A deaerator is placed at a height of about 20 Mts above B.F. Pp suction to avoid flashing and cavitation during a rapid load drop.

During a rapid load drop pressure in the heaters and deaerator tends to drop. This causes flashing in the deaerator as the water is stored at boiling point, corresponding to the pressure at which the water in the feed pump suction pipe gains static head as it descends must be greater than rate of pressure decay in the deaerator, if flashing in the pump is to be avoided. The suction pipe should be as near vertical as possible to avoid unnecessary head loss due to friction. Fig. illustrates the variation in pressure in a deaerator and at a feed pump positioned beneath it following a load rejection. The curve of pump suction pressure is greater, by a constant figure than the curve of deaerator pressure; this constant difference is equal to the static head of the deaerator on the pump. If the saturation pressure corresponding to the temperature of the water arriving at the pump is greater than the total pressure at the pump inlet, in other words static head plus deaerator pressure at that time less friction loss, then flashing will take pace in the pump. Because of static head, this situation is avoided. In modern units, a booster pp-is located before the main feed pump to further increase the feed – pp-suction pressure above saturation pressure at feed pump suction. The Deaerator is shown in fig.4.10.

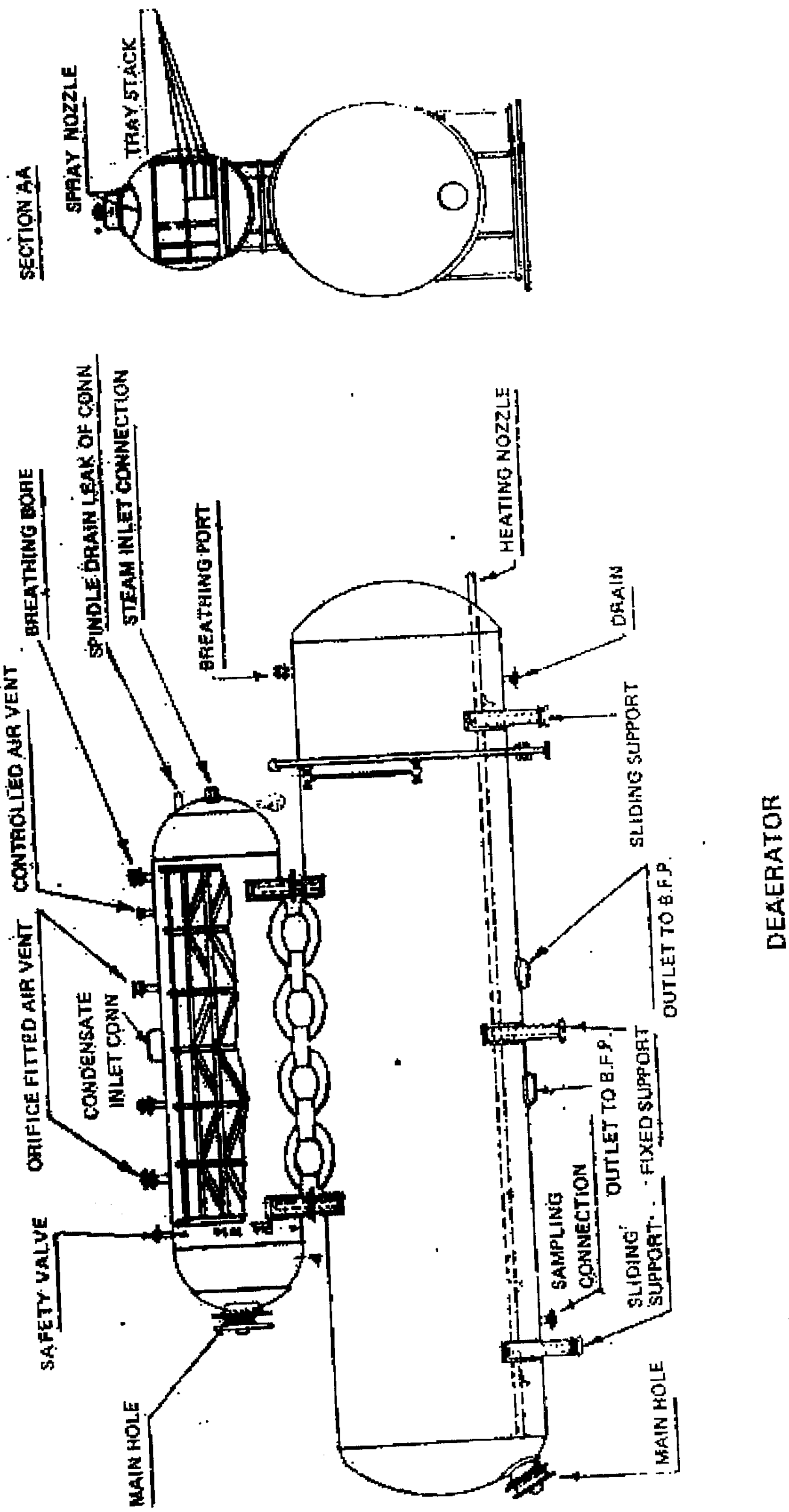


FIGURE 4.10 DEAERATOR

High Pressure Heaters

The feed water flows through the tube spirals and is heated by bied steam around the tubes in the shell of the heaters. These heaters are cylindrical vassal with welded dished ends and with integrated, desuperheating, condensing and sub-cooling sections. The internal tube system of spirals is welded to the inlet and outlet headers. As there are no flange ends the chances of tube leakages are less in this type of heaters. In order to facilitate assembly and disassembly, rollers at the side of the heater have been provided. Both feed water and steam entries and exists are from the bottom en d of the heater.

In 210 MW/LMW units, the feed water, after feed pump enters the HPHs 5, 6 & 7. The steam is supplied to these heaters form the bleed point Nos.3, 2 &1 of turbine through motor operated valves. These heaters have a group bypass protection on the feed water side, in the event of tube rupture in any of HPHs and the level of the condensate rising to dangerous level, the group protection device diverts automatically the feed water directly to boiler, thus bypassing all the 3 H.P. heaters.

The condensate of the bled steam formed in the heater is thrown either to the next lower stage heater in cascade or to the deaerator through a set of inter-locked valves depending upon the pressure condition inside the heaters. There is also an arrangement to take out air steam mixture form each heater in cascade and air steam mixture form each heater in cascade and air steam mixture is thrown to the condenser through the LP heaters.

4.10 TURBINE OIL SYSTEM

4.10.1 Purpose of Oil System

The turbine oil system fulfils four functions. It

a) Provides a supply of oil to the journal bearings to give an oil wedge at the shaft rotates.
b) Maintains the temperature of the turbine bearings constant at the required level. The oil does this by removing the heat which is produced by the shaft conduction, the surface friction and the turbulence set up in the oil.
c) Provided a medium for hydraulically operating the governor gear and controlling the steam admission valves.
d) Provides for hydrogen-cooled generators a sealing medium to prevent hydrogen leaking out along the shaft

It is worth noting that for 500 MW units and above, it is becoming the practice to use fire resistant fluids in place of lubricating oil for the control of governor gear and steam admission valves. These eliminate the risk of fire caused by leakage which is particularly likely when higher fluid pressures are used.

The schematic of the lubricating oil system for 210 MW (BHEL/LMW) turbine. Lubricating oil systems for power station steam turbines of other make or ratings also are a more or less similar.

4.10.2 Oil Specification (210 MW/LMW Turbine oil)

1. Recommended oil
   a. Turbine oil 14
   b. Mobil DTE medium

2. a. Specific gravity at 50^0C	0.852
b. Kinematic viscosity at 50^0C	28 cs
c. Neutralisation number	0.2
d. Flash point	201^0C (min)
e. Pour point	-6.6^0C (max)
f. Ash percentage by weight	0.01%
g. Mechanical impurities	Nil

System

The turbine oil system consist of the following

1. Main oil pump
2. Starting oil pump
3. A.C. Lub oil pump
4. D.C. emergency oil pump
5. Oil tank
6. Drain valve
7. Oil pressure drop relay (OPDR)
8. Oil Coolers

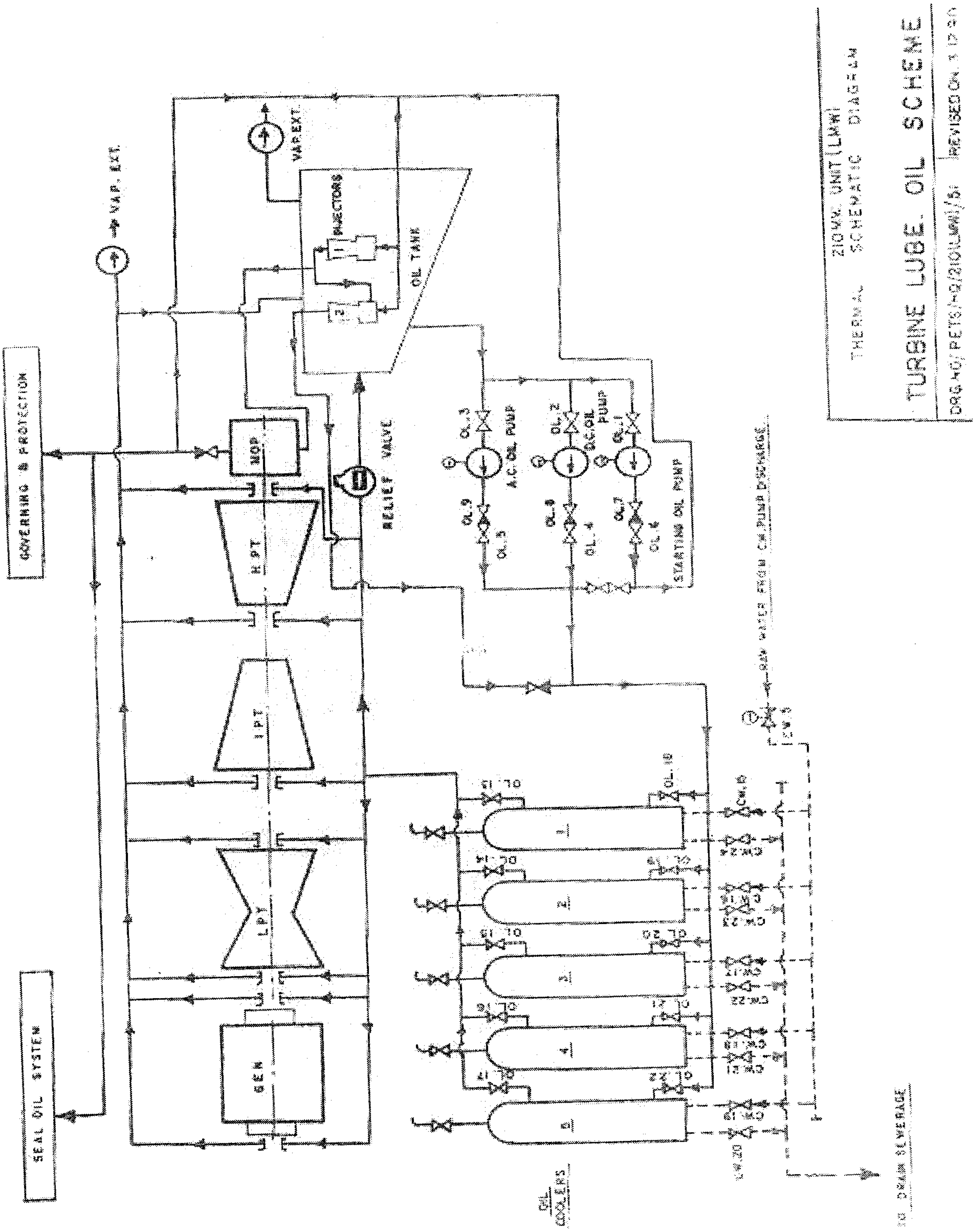


FIGURE 4.11 TURBINE OIL SYSTEM

Lubricating oil is supplied from oil tank (Capacity 28,000 Litres) to bearings and governing system with the help of pumps. The Turbine Oil system is shown in Fig.4.11.

During period of normal operation the required oil is supplied through the main oil pump mounted on the turbine shaft. A portion of discharge of the main oil pump is used as the working oil for the injectors. In fact there are two injectors located in the oil tank. The first injector supplies oil to the suction of the main oil pump and the discharged oil is discharged oil is further pressurized through the second injector which supplies oil to the bearings through coolers.

During initial starting a A.C. driven starting oil pump meets the requirement of both the bearing oil and governing oil.

Two standby oil pumps are incorporated in the system to supply bearing oil in emergency. One of these is A.C. driven and the other is D.C. driven.

4.10.3 Main Oil Pump

This pump is mounted in the front bearing pedestal. It is coupled with turbine rotor through a gear coupling. When the turbine is running tat normal speed i.e. 3000 rpm or the turbine speed is more than 2800 rpm, then the desired quantity of oil to the governing system at 20 Kg/cm^2 (gauge) and to the lubrication system at 1 kg/cm^2 (gauge) is the supplied by this oil pump. The oil to the lubrication system at the level of turbine axis is supplied through two injectors arranged in series. First injector develops a pressure of 3 kg/cm^2 (gauge) before oil coolers. After the oil coolers, the oil pressure is 1 kg/cm^2 (gauge) which goes to lubrication system. Main oil pump of a 500 MW unit. It is similar to main oil pump of 210 MW unit.

4.10.4 Starting Oil Pump (Auxiliary Oil Pump)

It is a multi-stage centrifugal oil pump driven by A.C. electric motor. Starting oil pump is provided for meeting the requirement of oil of the turbo set during starting. During starting or when the turbine is running at a speed lower then 2800 rpm it supplies oil to governing system as well as to the lubrication system.

4.10.5 A.C. Lub Oil Pump

This is a centrifugal pump, driven by an A.C. electric motor. This runs for about 10 minutes in the beginning to remove air from the governing system and to fill the oil system with the oil. This pump automatically takes over under inter lock action whenever the oil pressure in lubrication system falls to 0.6 kg/cm^2 (gauge). Thus this pump can meet the requirement of lubrication system under emergency conditions.

4.10.6 D.C. Emergency Oil Pump

This is a centrifugal pump, driven by D.C. electric motor. This pump has been provided as a back-up protection to A.C. driven lub oil pump. This automatically cuts in whenever there is failure of A.C. supply at power station and or the pressure in the lubrication falls to 0.5 kg/cm^2 (gauge).

4.10.7 Oil Tank

The oil is stored in oil tank of 28000 litres capacity upto operating level of the tank. About 4000 lit/min. oil remains in circulation. Liberally sized tank holds the oil inside the tank for a period long enough to ensure liberation of air from the oil. Different mesh sized fitters are located inside the tank to filter the oil during its normal course. The filters are easily accessible and removable for cleaning even when turbine is in service. This oil tank is supported on the framed structure just below the turbine floor at the left side of the turbine.

4.10.8 Relief/Drain Valve

This valve is mounted on the pipe line to maintain 1 kg/cm^2 (gauge) pressure at bearing axis in the lubrication system. So, whenever the pressure in lubrication system differs from the above value, the drain valve increases or decreases the oil drain to the tank and thereby maintain the required oil pressure in the bearing lubrication system.

4.10.9 Oil Pressure Drop Relay (OPDR)

The A.C. lub. Oil pump emergency oil pump come in service automatically under interlock action. The impulse for bringing the pumps into service is provided by oil pressure drop relay. It is possible to over ride the interlock manually to bring both pumps in service.
This OPDR comes into action under following conditions.

a) It switches on the A.C. lub. Oil pump when the pressure in the lubricating line drops to 0.6 kg/cm^2 (gauge)

b) It switches on the emergency oil pump when the pressure in the lubricating line drops to 0.5 kg/cm^2 (gauge)

c) It trips the turbine and prevents the operation of barring gear when lubricating oil pressure drops to 0.3 kg/cm^2 (gauge)

The relay is provided with a drain valve and an isolating valve for testing the reliability of its operation, even when the turbine is running.

CHAPTER -5

CALCULATION OF EFFICIENCY

5.1 CALCULATION OF BOILER, TURBINE , GENERATOR, AND OVER ALL EFFICIENCY OF UNIT –II :

5.1.1 Boiler Efficiency

Parameter taken for calculation :-

1) Total Coal Flow = 158.25 T/Hr.
2) Calorific Value = 3,390.00 kcal.
3) Main steam pressure = 135.00 ksc.
4) Main steam Temperature = 540.00^{0}C
5) Cold Re-heat Pressure = 24.60 Ksc
6) Cold Re-heat Pressure = 320.00^{0}C
7) HRH. Temperature = 535.00^{0}C
8) HRH Pressure = 23.80 KSC
9) Feed Water Temperature. at Eco.I/L = 242.00^{0}C
10) Boiler O/L Steam flow = 671.00 T/Hr
11) Main steam Flow boiler O/L flow – Aux Consumption
= 671-17 = 654.00 T/Hr
12) HRH/CRH flow = M.s Flow – (Ext. No. 1+Ext.No.2)flow
=654-(31.34+45.86)=576.80 T/Hr
13) Enthalpy of Main steam at 135KSC = 825.00Kcal/Kg
14) Enthalpy of CRH .steam at 24.6 = 730.00 Kcal/Kg
15) Enthalpy of HRH . .steam at23.8 KSC . and 535 ^{0}C = 848.00 Kcal/Kg

(Enthalpy Readings are Taken from NOLLIER CHART)
Ext.No.1and Ext.No.2 flow calculation in the Turbine Efficiency)
Calorific value from DM.Planet)

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(1) Heat gain in Main Steam / Kg =
= Enthalpy of Main Steam - Enthalpy of Feed water at Economiser Inlet

= 825-242 =583

Heat gain in Main Steam for 671 T.= 671 X 583 X 10^3
=3,91,193 X 10^3 Kcal/Kg .

---

(3) Heat gain in HRH = (Enthalpy of HRH .Steam -Enthalpy of CRH . Steam)
= 848-730 =118 K cal/Kg

ie. Heat gain for 576.8 T= 118 X 576.8 X 10^3
= 68062.4 X 10^3 Kcal/Kg

---

(4) Heat Output = Heat gain in Main Steam + Heat gain in HRH

$= 391193 \times 10^3 + 68062.4 \times 10^3$
$= 4592255.4 \times 10^3$ Kcal/Kg.

---(5)

Heat Input = Calorific Value of Coal X Coal flow (T/Hr)
$= 3390 \times 158.23 \times 10^3$
$= 536399.7 \times 10^3$ Kcal

---

(6)Boiler Efficiency =(Heat Output / Heat Input) X 100
$= (459255.4 \quad 10^3 \quad 536399.7 \times 10^3) \times 100$
= 85.618 %
ie. =85.6 %

5.1.2 Turbine Efficiency

(1) Parameters taken for Calculation :-

Ext No.1 (HPH-7)	Steam Pressure / Temperature	= 38.10 KSC /381 ^{0}C
Ext No.2 (HPH-6)	Steam Pressure / Temperature	= 24.60 KSC /319 ^{0}C
Ext No.3 (HPH-5)	Steam Pressure / Temperature	= 10.60 KSC /450 ^{0}C
Ext No.4 (LPH-4)	Steam Pressure / Temperature	= 5.60 KSC /390 .5 ^{0}C
Ext No.5 (LPH-3)	Steam Pressure / Temperature	= 1.80 KSC /268.0 ^{0}C
Ext No.6 (LPH-2)	Steam Pressure / Temperature	= 1.70 KSC /190.0 ^{0}C
Ext No.7 (LPH-1)	Steam Pressure / Temperature	= 0.28 KSC /110.0 ^{0}C

Feed Water flow	= 700 T/Hr
Condensate flow	= 555 T/Hr
Enthalpy of M.S. at 130 KSC / 540 ^{0}C	= 823 Kcal
Condenser Vaccum	= 0.11317 KSC (-675 mmHg)
L.P.T. Exhaus thood Temperature	= 47 ^{0}C

CONDENSATE TEMRERATURE :			DRIP TEMP.
LPH-1 Inlet / Outlet Temp .	= 50.4 / 58.3	^{0}C	56.80^{0}C
LPH-2 Inlet / Outlet Temp .	= 69.8 / 106.7	^{0}C	106.00^{0}C
LPH-3 Inlet / Outlet Temp .	= 106.7 / 126.5	^{0}C	127.00^{0}C
LPH-4 Inlet / Outlet Temp .	= 126.7 / 161.4	^{0}C	157.60^{0}C
HPH-5 Inlet / Outlet Temp .	= 165.0 / 182.0	^{0}C	168.50^{0}C
HPH-6 Inlet / Outlet Temp .	= 182.0 / 218.0	^{0}C	197.00^{0}C
HPH-7 Inlet / Outlet Temp .	= 218.0 / 242.0	^{0}C	291.00^{0}C
(G.C-2 Inlet / Outlet Temp.	= 61.0 / 69 .7	^{0}C	69.00^{0}C

ENTHALPY OF EXTRACTION STEAM :

Extraction No .1	=	757
Extraction No .2	=	730
Extraction No .3	=	805
Extraction No .4	=	775
Extraction No .5	=	723
Extraction No .6	=	667
Extraction No .7	=	645

Enthalpy of Feed Water is equivalent to its Temperature since Extraction flow at various stages – Direct readings not available these flow values are calculated based on "THERMODYNAMICS OPEN SYSTEM" (Assuming Perfect Insulation). According to this theory heat content in a open system constant.

Input Enthalpy = output Enthalpy.

Hence any LPH/HPH's inlet Steam / Water / Drip Enthalpy in total is equal to the outlet Drip / Water Enthalpy in total. The Turbine Extraction Diagram with its various heaters is shown in fig.5.1.

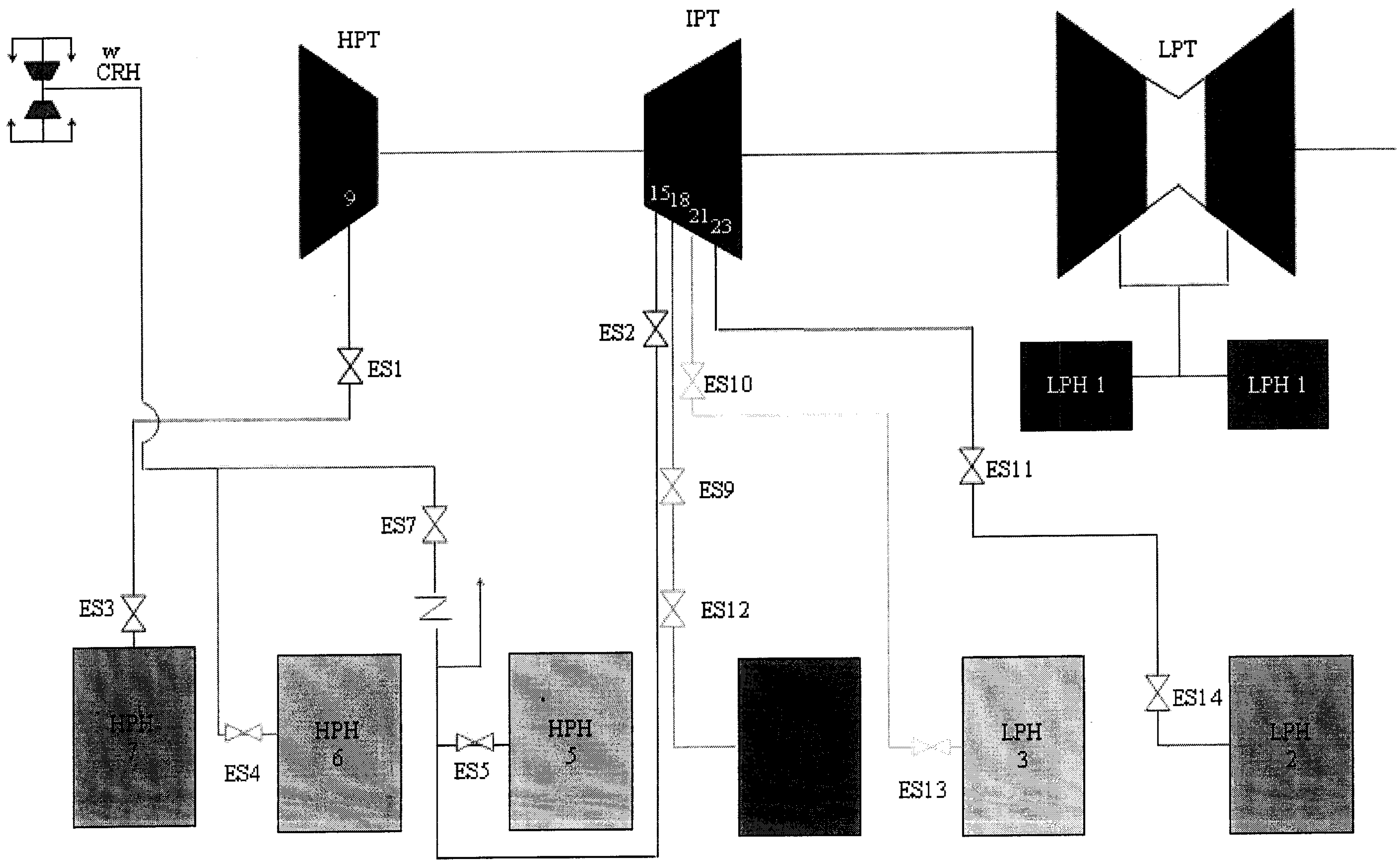


FIGURE 5.1 TURBINE EXTRACTIONS

(1)EXTRACTION STEAM FLOW CALCULATION (HPH-7)

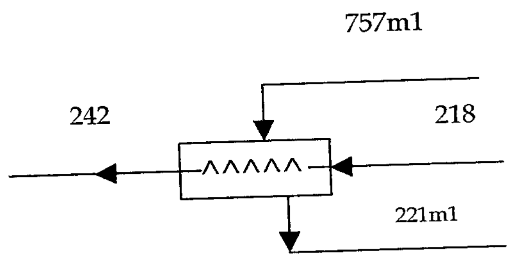
$= 757m1 + 242$

$= m1(757-221) = 242-218 = 24$

$m1 = 0.04477$ Kg / Kg of water.

ie .Extraction flow $= 0.04477 \times 700 \times 10^3$

$= 31.34$ T / Hr .



EXTRACTION STEAM FLOW HPH-6:

$= 730m2 + 182 + 221m1 = 218 + 197(m1 + m2)$

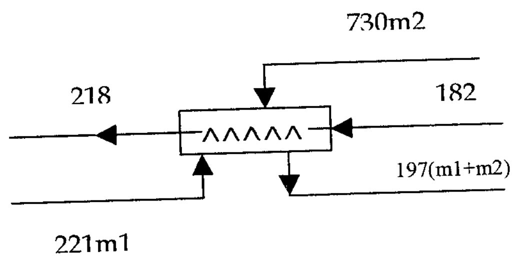
$= m2(730-197) + m2(221-197)$

$= 218 - 182 = 36$

ie . $m2 = 0.0655$ Kg /Kg of feed.

ie . Extraction Steam flow $= 0.0655 \times 700 \times 10^3$

$= 45.86$ T/Hr



EXTRACTION STEAM FLOW (HPH-5)

$804m3 + 165 + 197(m1+m2)$

$= 168.5(m1+m2+m3) + 182$

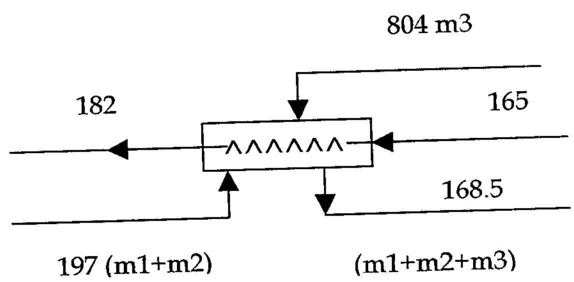
$= 804m3 - 168.5m3 + m2(197-168.5) +$
$m1(197-168.5)$

$= 182 - 165 = 17$

ie $m3 = 0.02177$ Kg/kg of feed.

ie. Extraction flow $= 0.02177 \times 700 \times 10^3$

$= 15.269$ T/Hr.



EXTRACTION STEAM FLOW LPH-4.

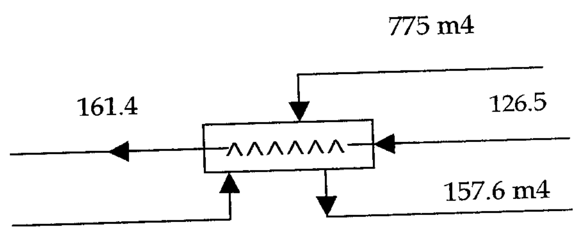
$= 775m4 + 126.5 = 161.4 + 157.6m4$

$= m4(775-157.6) = 161.4 - 126.5 = 34.9$

$= m4 = 0.05652$ Kg/kg of feed

ie. Extraction Steam flow $= 0.056652 \times 555$

$= 31.37$ T/Hr.



EXTRACTION STEAM FLOW LPH-3.

$= 723m5 + 106.6 + 157.6m4$

$= 126.5 + 127(m4+m5)$

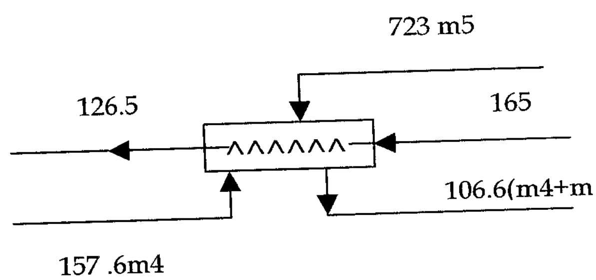
$= m5(723-127) + m4(157.6-127)$

$= 126.5 - 106.6 = 19.9$

$= m5 = 0.0304$ Kg/kg of feed

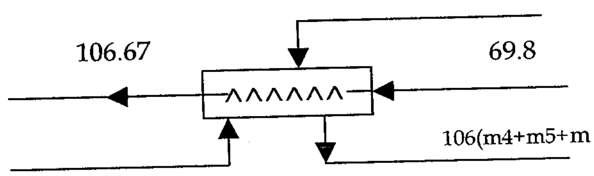
Extraction steam flow $= 0.0304 \times 504$

$= 16.92$ T/Hr.



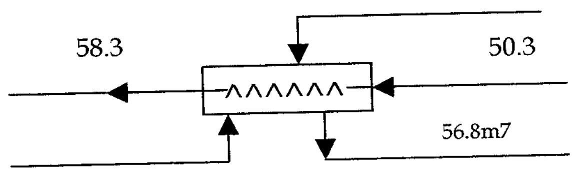
EXTRACTION STEAM FLOW LPH-2.

= 669m6+69.8+127(m4+m5)
= 106.67+106(m4+m5+m6)
= m6(669-106)+(m5+m4) (127-106)
= 106.67-69.8
=m6=0.06246 Kg/Kg of feed
ie. Extraction steam flow=0.06246X555
=34.66 T/Hr.



EXTRACTION STEAM FLOW LPH-1

=645m7+50.4= 58.3456.8m7
=m7(645-56.8)=58.3-50.4=7.9
m7=0.01343 Kg/Kg of feed.
ie. Extraction steam flow=0.01343X555
=7.454 T/Hr.



HRH. STEAM FLOW = Main Steam flow-Ext.(1+2) Flow
= 654 –(31.34+45.86) = 576.8 T/Hr.

STEAM FLOW TO L.P.T = HRH. flow-(Ext.3+Ext.4+Ext.5+Ext.6) Flow.
= 576.8-(15.269+31.37+16.92+34.66)
= 576.8-98.22=478.58 T/Hr.

STEAM FLOW TO CONDENSER=(Steam flow to LPT – Ext. Stm.flow to LPH-1)
=478.58 – 7.454 = 471.13 T/Hr.

WORKDONE IN H.P.T:

Main Steam flow (En. of M.S.-En. of Ext.No.1)+ (M.S.flow – Ext.No.1 Flow)
(En. of Ext.No.1 flow-En.of CRH) X 10^3
= 654(825-757)+(654-31.34) (757-730) X 10^3
=(44472+16811.82)10^3 = 61283.82 X 10^3 Kcal.

WORKDONE IN IPT.

HRH FLOW(En. of HRH- En. of Ext.No.3)+(HRH flow – Ext.No.3 flow)
(En. of Ext.No.3 – Ext.No.4)+(HRH. flow – Ext.No.3+4 flow)
(En. of Ext.No.4 – En. of Ext.No.5)+(HRH. flow – Ext. No.3+4+5)
(En. of Ext.No.5 – En. of IPT. Exht)

= 576.8(848-805)+(576.8-15.26)(805-775)+(576.8-15.26+31.37)
(775-723)+(576.8-15.26+31.37+16.92)(723-667) X 103
=576.8 X 43 + 561.54 X 30 + 530.17 X 52+513.25 X 56
=24802.4 + 16846.2 + 27568.84 + 28742.56
=24802.4 + 16846.2 + 27568.84 + 28742.56
=97960 Kcal.

WORK DONE IN LPT.
Steam flow to LPT(En. of Ipt Exhauset – En. of Ext. No.7)+(Steam flow to LPT – Ext.No.7 flow) (En. of Ext.No.7 – En. ofL.PT. Exhaust)

= 478.58(667-645)+(478.58-7.454)(645_615)
= 478.58 X 22 + 30 X 471.13
=10588.76 + 14133.9
= 24662.66 X 10^3 Kcal.

Total work done = work done of (HPT+IPT+LPT)
= (61283.82 + 97960 + 24662.66) 10^3
= 183906.48 X 10^3 Kcal.

HPT. IN PUT TURBINE = 459926.4 X10^3 Kcal.

ie. Efficiency of Turbine $= \dfrac{\text{work done in Turbine}}{\text{Heat input llto Turbine}} \times 100$

$= \dfrac{183906.48 \times 10^3}{459926.4 \times 10^3} \times 100$ =39.99%

5.1.3 Generator Efficiency:-

Generator output =210MW=210 X 10^3 KW.
Heat equivalent of 1 KW =859.9 Kcal
@ Heat equivalent of 210 X 10^3 KW.
=210 x 10^3 X 859.9 Kcal/x

ie. Efficiency of Generator= $\dfrac{210 \times 10^3 \times 859.9}{183906.48 \times 10^3} \times 100$

=98.19%

5.1.4 Overall Efficiency

Efficiency of (Boiler X Turbine X Generator) X 100%
=0.85618 X 0.3999 X 0.9819
=33.62%

DIAGRAMMATIC HEAR BALANCE OF 210 MW. UNIT-II

1) Total Heat input =Coal flow X Calorific Value
=158.23 X 3390 X 10^3

= 536399.7 X 103 Kcal/hr.

Loss in Boiler with respect to heat input

$$\frac{(536399.7 - 459926.4)\ 10^3}{536399.7 \times 10^3} \times 100$$

=76473.3/536399.7 X 100 = 14.257%

2) Generator output with respect to Total Heat input

$$= \frac{210 \times 10^3 \times 859.9}{536399.7 \times 10^3} \times 100$$

= 33.665%

3) Loss in Generator with to heat input

$$\frac{\text{Workdone in turbine-Gen.output}}{\text{Heat input.}}$$

$$=\frac{(183906.48 - 180579)10^3}{536399.7 \times 10^3} \times 100$$

$$=\frac{(183906.48 - 180579)10^3}{536399.7 \times 10^3} \times 100 = 3327.48$$

=0.620%

4) Heat loss in Turbine with respect to total heat input.

$$\frac{\text{=Heatinput lto turbine-(workdone in Turbine + loss in Condenser)}}{\text{Heatinput}}$$

$$= \frac{459926.4 \times 10^3 = -(183906.48 \times 10^3 + 268072.97 \times 10^3)}{536399.7 \times 10^3} \times 100$$

$$= \frac{459926.4 \times 10^3 - 451979.45 \times 10^3}{536399.7 \times 10^3} \times 100$$

= 1.4815% ie. =1.482%

5) Heat loss in condenser with respect to total heat = $\dfrac{\text{En. of Steam at condenser inlet – En. of inlet – En. of Condensate at Hotwell}}{\text{Heat input}}$

$$= \frac{(615-46)10^3 X\ 471.13}{536399.7\ X\ 10^3}\ X100$$

$$=(268072.97/536399.7)\ 100=49.976\%$$

CHAPTER – 6

PERFORMANCE MONITORING OF TURBINE CONDENSER AND FEED WATER HEATERS :-

The recent jump in power demands had given sharp impetus to install generating units of higher capacities 210 MW and 500 MW. The cost of power station erection has increased and with the inflationary trends there is always on escalation. This necessitates higher selling rates which in the case of power is likely to be not permitted to be raised very high due to various pressures on the generating station. These conditions demand economic inability of the plant for its operational cost and installation cost requiring high availability, plant load factors and efficiency. When the efficiency and reliability demanded is high it becomes important to know, measure and control where these are likely to deterioate. This is carried out by monitoring the performance of the equipments concerned along with the systems involved. The aspects that relate to Turbine, condenser and feed heaters will be dealt below :

6.1 TURBINE PERFORMANCE MONITORING :-

It is imperative to know the losses that are associated with turbine and their reasons before the monitoring system is developed. Turbine losses are divided into internal and external. The internal losses are due to blade and disc friction. leakage, wetness, back to leakage, bearing and heat radiation. Based on the above losses and their progressive deterioration. the performance of the turbine will deterioate. Thus by monitoring the performance on a regular basis the condition of the turbine could beb predicted and necessary steps could be taken.Hence the points of parameters to be monited are to be determined as a first step.The two major parameters are the pressure and temperature.For a given pressure at turbine stop valve the internal pressures vary linearly with load.Hence the pressure survey is the prominent monitering parameter.

In the pressure survey technique the pressures of TSV,first stage or impulse stage ,HP exhause,IP inlet ,LP inlet and every extraction or regularly noted for a fixed load and comparred with design and initial performance test values.The pressures are expected to be more as the internal leakage is increased, the pressures are expected to be less if there is a blockage or increase in internal friction.As per the result of the latest pressure survey it is possible to point out what sort of deterioation had taken place.

If any deviation is noted in the pressure survey diagram the reason could be interpreted as internal blackage od due to silica deposits.Any internal blockage occurs suddently wherev as silica deposit affects gradually.Another common cause of loss of performance is that of blade roughness.A sligh blade roughening is sufficient to cause a significant change of performance.Roughness of only 10 is sufficint to cause the heat rate of 200 MW machine to worsen by 1%.

From the above , it is seen that the pressure survey of turbine is the important monitoring system for turbine.

The next losses are due to gland leakage due to wear in the seals. The leakages through glands of turbine and steam valves are lead to gland steam condencer. Hence the monitoring of the gland steam temperature, flow in gland steam cooler and temperature raise in condensate in gland in staewm condencer will give the sign of wear. Hence the loss as well as the degree of wear could be approximated from this monitoring so that effective measures could be taken during the overhauling of the Turbine.

The loss on radiation could be monitoring through a measurement of surface temparature of the Turbine insulation and the extraction pipes. These of course could be corrected only during a major overhaul.

The monitoring techniques described so far were on long term basis while there are certain parameters which are to be monitored on shift basis to avoid deviating from the optimum conditions. These are TSV staem pressure,TSV and reheat steam temparature ,Reheater spray , final speed water temparature and condencer back pressure .

Varying the turbine inlet pressure causes :-

- The heat drop of the steam to vary, higher pressures causing increased heat drop.
- Increasing exhaust wetness and higher pressures.
- Increased turbine output for a given value opening such that 10% increase pressure will produce about 10% extra out put.

With throttle governed turbine the output is proportional to the pressure after throttle value which inturn depends on pressure at turbine stop value. The increase in pressure gives increased output and reduced heat rate which implies that the efficiency will improve. But with increase in pressure the system is very much affected , as the steam flow increase, the condencer pump output and boiler feed pump output have to be increased. As the extraction pressures also increase thefeed temperature after heaters increarse. As the flow to condencer increases the back pressure also is effected. However with pressure drop at TSV the cycle efficiency is adversely affected. However operating the throttle governed turbine at partial loads with reduced pressure with temperatutre not varied significantly improve the system by reduce wetness at the exhaust and saving ibn power in boiler feed pump to raise the head only to a lower level. However reduced pressure operation is adviceable only at loads lower than 50% in throttle governed turbines.

The temperatures at Turbine stop valve andb Reheat very much affect the heat drop and as the temperature is increased the heat drop increases and heat rate reduces. However it must be remembered that the material for turbine and tubes for super heaters and reheaters are selected with maximum permissible metal temperature very close to operating parameters and hence the life of components are reduced when operating at higher temperatures than prescribed optimum. The lower temperature will reduce the heat drop and hence the efficiency drops. The wetness of the steam at the turbine exchaust also will increase.

Hence monitoring the steam temperatures and pressures on shift basis will give an idea for the operating engineer how efficiently the machine was operated.

The next major parameter that affect the efficiency is the reheater spray for temperature control. The main reason for reducing of efficiency is that the reheater apray quantity does not give its work in HP turbine and that apart it increases the stage pressures in IP and LP and heat is supplied more in boiler as the spray water is taken before the nfeed heaters. Thus reheater spray increases the heat rate. Hence it becomes neccessary to monitor the reheat spray and efforts to aviod or to keep as minimum as possible are to be taken.

Feed water temperature after top heater is the other parameter to be monitored on shift basis. This will indicate the performance of the heaters and corrective steps are to be taken as quickly as possible such as maintaining correct levels, checking for by passing etc.

The next major parameter for good turbine performance is the back pressure at condencer. The deviations must be analysed in shift operation and corrective steps to be taken to aviod air increarse, to improve C.W . flow and opweration cooling towers.

The major parameters that affect turbine performance and these to be monitored were discused.There are certain other parameters though minor in nature will improve the operation of equipment. They are the turbine lube oil temperature, differential pressures across filters, governing oil pressure, chemical analysis of feed water,steam purity and silica in condensate.

The specified instruments for measuring pressure and temperature shall be calibrated once in three months and a record must be kept, so that when actually an incident acurrs there need not be any suspicion on instrument. Regular recording of readings in the aclal instrument fitted in syatem and recoding of readings with special calibrated instrment during tosts will also give the errors in the actual instruments. Hence the trend can be always noted. It is important that the readings taken shall be interpretted immediately and a remak is also recorded. the usage of Date acquisition system will give a great help in maintaining the readings and records.

6.2 MONTITORING OF CONDENSER PERFOMANCE :

The Condenser maintains the back pressure for the turbine to complete the expansion of the steam in Tubine to give its useful work. Even small improvement at the back pressure gives additonal output. Hence monitoring condenser performance is one imporlant task of any power plant. As there advatage in improing the back pressures there are also losses due to increased leaving losses, increased wetness of steam and reduced condensate temp. Hence for each load an optimum back pressure must be maintained beyond which the losses overtake the advantages.

The factors that affect the condenser performance are :-

1) Circulating water inlet temperature

2) Circulating water flow

3) Air into syslem

4) Dirty or foulee tubu

Circulating water inlet temperature play a signifcant role in maintaining the back pressure. All the condensers are designed with cartain given C.W. inlet temperature which is approximately the maximum temperature expected at the location of the power station during summer. mowever there will be variations throughout the year as well as every hour in a day. A family curves for different C.W intlet temperature with the conndenser is first intalled (clean) must be taken and recorded such that these can be used as a reference. The C.W temperature can be controlled by additional cooling towers or by increasing the number of indraft fans if common bus system for C.W. pumps is used. when unit system is used this will involve additional power for addition C.W. power to be put into service. Another control is to increase the circulating water flow throughthe condenserif the system provides for such arrangement and reduce the C.W. out let temperature so that the lower C.W. temperature is obtained also will give information as to whether steam input to comdenser is as per the designed optimum. This is more important because many drains are connected to the flash tank and the heater dreins are also connected. Any fault (open position of valve instead of close position) in the system or passing of valves will increase the steam ioad on the condenser which goes to increase the C.W.out let.

Circulating water flow infueacess the back precasure by reducing or increasing the heat transfer. The out let temperature increases and the temperature difference also will increase. This can be corrected only if system iss so arranged that additional C.W. be started. otherwise the reason for reduetion of fiow is to be thoroughly checked.However it most be remebered that increasing flow above a certain limit would erode the tubes at the entry point and the velocity of the cooling water dose not excced the design limit of about 2m/s.

The next condition is the air increases into vacuum system. This can be detected by higher exhaust hood temperature when other parameters are normal. The difference between exhaust hood and hotwell condensate will increase. If additional air ejectors are put into service the vacuum will improve fast. The entire systems are to be checked patiently to detect the air leakege. Additionally when unit is nit is out of service, water can be filled up on steam side upto condenser joint to LP turbine and check for water leak. If necessary fluroscent powder can be mixed in the even very light leak. Air ingress will very seriously affect the heat transfer by forming an insulating medium over the tube surface.

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The following factors also may affect the heat transfer across condenser tubes to a lessor extent :

- Resistance due to the condensation film on the steam side of the tube.
- Deposits such as copper oxide. on the steamside of the tube.
- The resistance of the tube material

The stagnant water film adjacent to the inside of the tubes.

From the above desoription of various parameters that affect the condenser the parameters to be monitored for condenser performance could be easily derived. They are

1) Load on the machine

2) Condenser back pressure

3) Circulating water inlet temperature

4) Circulating water outlet temperature

5) Exhaust hood temperature

6) Condensate temperature

7) Vacuum drop per min.

Trend graphs of continous Fall in vacuum as the running hours increase would give an idea how the vacumm is expected to behave in the future.

6.3 MONITORING OF FEED WATER HEATER PERFORMANCE :

The regenerative system increases the efficiency of the cysle and hence any fall in the performance of the feed geaters will very adversely affect the system over all efficiency.

The performance of heaters can deteriorate due to the

following :

a) Partial by - passing of heater

b) Water side scaling

c) Steam side scaling

d) Water level in heater

e) Air removal system not working

The major reason for heater performance loss during sgift operation can be dur to by - passing and not maintaining proper level. Eventhough the bled steam in the particular heater which is by - passed on the water side remains normal the outlet water temperatre becames less and when thes passes through the mext henter it draws more steam from the turbine to heat the water from a temperature lower than normal to the saturation temperature of heater.

Thes fobs the equivalent output that could be got from this quantity of steam if allowed normally thorugh Thubine. When the level in the heater is not maintained properly the Terminal temperature difference increases if level is very low or no level and it floods the condensing tube surface if hte level is very high.

The next major reson for heater performance loss is due to air lock and non-condensing air/gas not removed properly. It is very imporatant that during commissioning of heaters when start up of unit the air vents are properly opened and the cetir air is remover through condenser and during service also the air valve for condenser is kept open. during priming of heater air is to be removed from the top most poirt of heater where as during operation air is to be removed at the farthest end in condensing zone frfom steam inlet. It is important in the shsift to check the valves for proper open and close position. Air blanketing of the heater tube surface will adversely affect the heat transfer.

The other reasons for performance loss is due to contaminaton on tube surface from inside as well as from out side. These are very radual or sudden also. The silica mixing in the gotwell due to condenser tube puncture will lead to deposits on the water side of the tube. Also oil escape int hotwell from LP bearings will also leave a light oil deposit on the tube surface on water side. Suden collapse of baffles or flaking of surface material on the tube side will adversely affectthe heat transfer.

The performance of heater could be monitoredmainly be the terminal temperature difference an the steam pressure periodical test for heaters gcat balance would reveala the performance level

of the geaters. whenever a tube puncture is detected the heater must be put into service as quickly as possible after attending to the defect.

When the heater is out of service for maintenance works the immediate effect is the flow of steam through turbine at downstream stages increases and condenser back pressure also will increase. The next effcet is that since the water inlet temperature to the upstream heater will be low and this would increase the bled steam flow through it. However if the heater isolated is the top one then the final feed water temperature also is adversely affected bringing down the cysle efficiency.

7. CONCLUSION

The working of 210 MW steam turbine is studied here and the heat losses associated with the turbine is estimated by calculating the work done in various parts of turbine, the losses associated with various extractions that are available in the high pressure, Intermediate pressure and low pressure parts of the turbine.

Heat loss is estimated and overall efficiency of a single unit of thermal power station is calculated. Some of the suggestions for reducing the heat loss and improving the performance of the turbine is discussed

Future Work

In future the performance analysis of the steam turbine can be improved still by linking any software and to study the performance in improved way and also implementation of energy conserving methods and there by improvement of system performance can be studied.

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